COMBUSTION

Vol. 2, No. 3 SEPTEMBER 1930 25c. a copy

NEW YORK CITY FROM GRAND CENTRAL ZONE TO THE BATTERY

The Performance of Natural Draft Chimneys

By J. G. MINGLE

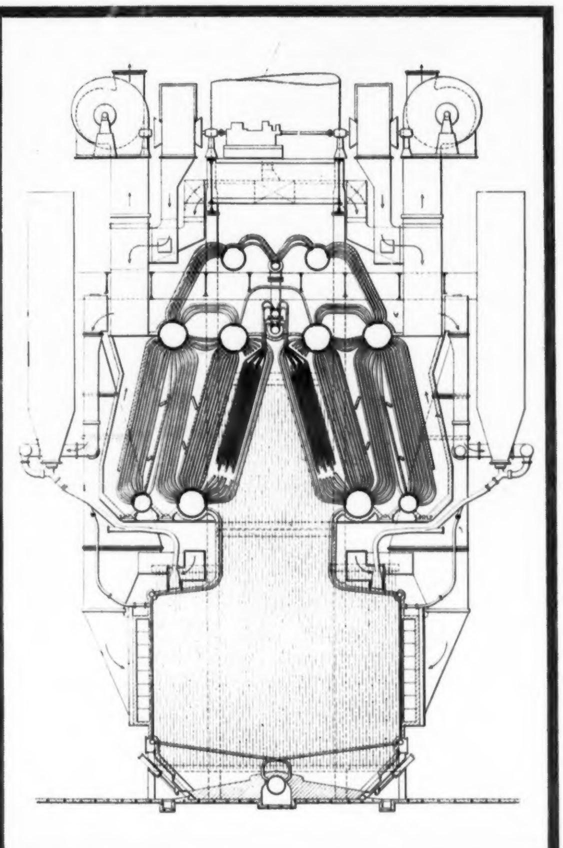
Direct Firing Steel Plant Boilers with Roller Mills

By HENRY KREISINGER and A. C. FOSTER

Other Articles in This Issue By

A. G. WIKOFF • WM. L. DE BAUFRE • DAVID BROWNLIE • E. J. PATTON • B. J. CROSS

A Million pounds of Steam per hour at 86.5% efficiency



HEAT BALANCE of
No. 7 BOILER OUTPUT 1,000,000 lb. per hr.

	B. t. u.	*Per cent*
Loss due to moisture in coal	11	0.1
Hydrogen	453	3.0
Dry chimney gases	1,137	7.7
Combustible in refuse	73	0.5
Moisture in air	30	0.2
Radiation and unaccounted for	294	2.0
Total losses	1,998	13.5
Efficiency and heat to boiler	12,772	86.5
Total	14,770	100.0

The above heat balance is from paper presented at the Summer convention of the A.I.E.E., Toronto, Ont., Canada, June 23-27, 1930,—by C. B. Grady, Mechanical Engineer, W. H. Lawrence, Chief Operating Engineer, and R. H. Tapscott, Electrical Engineer, of The New York Edison Company.

The coal used in test of No. 7 Boiler:
Proximate Analysis (dry basis)

Volatile	21.5 per cent
Fixed Carbon	72.4 per cent
Ash	6.1 per cent
B.t.u.	14,770.
Sulphur	1.4 per cent

(Moisture in coal as fired—1%)

Boiler unit No. 7 is one of three Combustion Engineering Steam Generating units installed in the East River Station of The New York Edison Company.
This heat balance shows the results of a twelve hour test run, where evaporation averaged 1,000,000 lb. per hour. For peaks, this unit has operated at the rate of 1,270,000 lb. per hour.

COMBUSTION ENGINEERING CORPORATION

200 Madison Avenue **New York, N. Y.**

BOILERS - AIR PREHEATERS - STOKERS - PULVERIZED FUEL EQUIPMENT - WATER-COOLED FURNACES

// COMBUSTION

VOLUME TWO • NUMBER THREE

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COMBUSTION

Vol. 2 September 1930 No. 3

Boiler Manufacturing Technique is Keeping Abreast of the Times

HAYS H. CLEMENS

DURING the past two or three decades changes in product, merchandising, and technique have greatly altered the character of the boiler business.

Many of us can remember when a boiler shop built large numbers of small units, simple enough in design to permit carrying on hand large stocks of fabricated boilers. The accessory equipment manufactured was usually restricted to grates, stacks, and occasionally included small steam engines.

The electrification of industry and the obvious economy of concentration have resulted in the elimination of a large part of the small boiler business, which has been supplanted by the manufacture of a smaller number of relatively large units, often specially designed to the needs of a particular installation. Many manufacturers have also found it necessary to add to their line of products, by development or association, combustion equipment of improved efficiency, and many other accessories formerly of independent origin.

The majority of boilers are no longer sold off the shelf. The sale of a large boiler installation frequently involves a great deal of engineering, proposals in exhaustive detail, and the assumption of broad responsibilities formerly unknown to the industry.

In the field of technique, shop practice has undergone radical changes. The old punch and bevel shears have been supplanted by the high speed drill and the planer; high pressures have demanded heavier plate, modern equipment, close tolerances and machined fits; the use of fusion welding increases daily, and has practically eliminated riveting in low pressure boilers.

The boiler industry has not only kept abreast of the demand for larger units, higher economy, and superior performance, but has often led the way to higher efficiencies and new accomplishments that would have been considered fantastic or impossible until recently. These successes have been achieved by a determined and continuous attack on the problem of steam generation and steam generating equipment. A high degree of engineering skill has been developed in the course of this attack, and research by engineers, metallurgists, chemists, and other specialists has contributed largely to its success. The industry today is alert, well equipped, and steadily pushing forward the improvement of its product.

This high degree of development has been arrived at in the face of practically a fixed market. Any increase in demand for steam generating equipment has been more than absorbed by the increased capacity and efficiency of the present day product. The boiler industry as a whole may congratulate itself upon having made invaluable and often unrewarded contributions to the mechanism of modern civilization, and upon a state of fitness of equipment and technical facilities which insures the continuance of those contributions.

Hays H. Clemens

President, American Boiler Mfg. Association

EDITORIAL

The Coal Industry Shirks

THE coal industry is, in many respects, a giant. However, when it comes to fighting its own battles its record is not impressive.

Some years ago, when hydro-power and fuel oil challenged the supremacy of coal in the field of power generation, coal figuratively sat down and waited for conditions to improve.

More recently, fuel oil and natural gas have cut deeper each successive year into the tonnage that coal long enjoyed in the domestic market. Coal sadly counted up its losses and ventured the thought that—"something should be done."

While coal sat and waited, during the competition in the steam plant field, the manufacturers of boilers, stokers, pulverized fuel apparatus and furnace equipment waged and won the battle for coal, by raising the standards of coal burning to levels which retained its supremacy.

Conditions improved and coal passed a crisis.

In the present competition between coal, and oil and gas for supremacy in the domestic fuel field, the manufacturers of small stoker equipment are bearing the brunt of the battle as far as coal is concerned.

Fortunate indeed is the coal industry that it has such worthy contemporaries to champion its cause. The proverbial race between the hare and the tortoise might be repeated in reality were it not for the continued alertness, energy and ability of the equipment manufacturers who have steadfastly protected and furthered the interests of the coal producers.

Certain individual coal producers have recognized the need of extending their interests beyond the mere mining and loading of coal and have employed fuel experts and combustion engineers to assist the ultimate consumer in the proper selection of coal and in its efficient utilization.

While this is unquestionably a step in the right direction, it is principally a defensive move against shrinking markets.

The problem still towers above these isolated attempts at its solution.

The coal industry seems to lack unity of purpose. Each producer faces the problem alone and seeks as best he can for a solution.

The attitude of the industry is largely individualistic.

The co-operative spirit that characterizes so many of our more successful industries is sadly missing.

The coal industry in its entirety should join more actively in the work of maintaining and extending its markets, by better mining and cleaning methods, by closer cooperation with equipment manufacturers in research and development, and by proper education of the ultimate consumer in efficient coal burning methods.

This task of stabilizing its markets is a responsibility that the coal industry cannot afford to delegate or disregard.

Long on Wealth, Short on Courage

IN a recent survey, the Harriman National Bank and Trust Company made an excellent presentation of the present status of physical and mental conditions in America.

With 7 per cent of the world's population, the United States consumes 48 per cent of the world's coffee, 53 per cent of its tin, 56 per cent of its rubber, 21 per cent of its sugar, 72 per cent of its silk, 36 per cent of its coal, 42 per cent of its pig iron, 47 per cent of its copper, 69 per cent of its crude petroleum, and over 23,000,000 of the 30,000,000 running automobiles.

It operates 60 per cent of the world's telephone and telegraph facilities, 33 per cent of the world's railroads, and produces and consumes more than 35 per cent of the world's total electric power.

This nation embraces 6 per cent of the world's area, but it produces 70 per cent of all the oil, 60 per cent of the wheat and cotton, 50 per cent of copper and pig iron, and 40 per cent of the lead and coal output of the globe.

It holds about one-half of the world's monetary gold and two-thirds of the total banking resources of the earth.

While its population was increasing 60 per cent, its industrial production increased by 300 per cent. The purchasing power of the 120,000,000 citizens is greater than that of the 500,000,000 Europeans, and is much greater than that of the more than a billion Asiatics.

On the other hand, we would seem, by the pessimistic sentiment prevailing, to have about 1 per cent of the courage, three-quarters of 1 per cent of the nerve, one-half of 1 per cent of force and power, and one-quarter of 1 per cent of backbone of almost any one country—England, for instance—struggling along, carrying gigantic debts and with millions of unemployed, without a murmur or complaint.

If American business will lay aside its blue glasses and review conditions in the full light of day, our return to normalcy will be swift and certain.

The Performance of Natural Draft Chimneys*

By J. G. MINGLE, Indianapolis, Ind.

This discussion of natural draft chimney performance is made particularly interesting and understandable by comparison with the performance of a centrifugal pump. The author has found that there is a rather surprising degree of similarity between the readily grasped principles of centrifugal pump performance and that of a natural draft chimney, the functioning of which is frequently misunderstood even by those intimately identified with steam plant operation. The analogy is nicely developed, both in the text of the article and in the accompanying curves. In later articles to be published in COMBUSTION, Mr. Mingle will discuss the economical design of natural draft chimneys and their general characteristics.

•

ONE of the most important and difficult problems encountered in the operation of the present day steam power plant isth at of bringing a definite quantity of air in contact with a given weight of fuel so that combustion will be as complete and perfect as possible. The ideal condition is that which permits of a constant relation between weight of fuel burned and quantity of air required *at all rates of combustion*. In practice, however, such a theoretical union is not only impossible to obtain but is not even approached closely due to the fact that the ordinary physical state of most fuels does not permit of such a union. In powdered fuel installations only do conditions approaching the ideal exist for a perfect union of air and fuel. As a general rule, however, in the average plant, fuels must be accepted and burned in their ordinary commercial state and in order to burn them properly and efficiently the air must be fed to them by an "air-feeder" which should be as efficient and economical in operation as possible.

Air is fed to the fuel during the process of combustion by means of a draft producing system. The production and maintenance of draft is of unsurpassed importance in the design and operation of the steam producing plant. No part of the entire plant has a greater influence upon its smooth and efficient operation than the draft producing system. The effects of draft may be considered as the "breath" and the draft producing system the "lungs" of the plant. Just as in the human system, any defects showing up in the "lungs" will react immediately upon the rest of the plant with disastrous and costly results. It is, therefore, of the greatest importance that the draft producing system be not only generally adaptable to the conditions which it is to serve but also that it be of the most efficient and economical type.

Contrary to a general impression, draft is not a difficult subject to understand. Draft in a steam plant is analogous to head in a water works system and the general theory of draft may be developed along the same line of reasoning as the general theory of hydraulics. Static head, loss of head and dynamic head in a water works system, for example, are analogous to static draft, loss of draft and dynamic draft in a draft producing system. Draft is associated with gases and head with liquids, generally water.

Draft and head, in general, may be defined as a pressure differential. In order to create this pressure differential, there must be, in both cases, an agitating mechanism. This agitating mechanism may be termed a "pressure transformer." In the case of water, the most common pressure transformer is a centrifugal pump. In the case of gases, there are three general types of pressure transformers, viz: (1) Induced draft and Forced draft Fans, (2) Venturi chimney, and (3) Natural draft chimney.

These three general types of pressure transformers naturally relate to the four general types of draft producing systems, viz: (1) Induced draft systems, (2) Forced draft systems, (3) Venturi draft systems, and (4) Natural draft systems. It is with Natural draft systems that this article will deal.

The essential element of a natural draft system is, of course, the natural draft chimney. The natural draft system of draft production is the oldest system in use and, until comparatively recent years, was practically the only type employed in steam plants. While natural draft chimneys have been used for hundreds of years in connection with various other types of installations, their extended use and application in steam plants have been developed to a great extent only since the wide application of the modern steam boiler. Despite the fact that natural draft systems possess certain disadvantages which militate against their extended use under all conditions,yet the great majority of the plants operating today employ this system of draft production either wholly or in part.

A typical natural draft system consists essentially of a relatively tall natural draft chimney built of steel, brick or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all of the heat has not been extracted. Hot gases are an essential element in the operation of natural draft systems, whereas in the other draft producing systems the gas temperature plays no active part. The results obtained by natural draft systems are identical with those obtained by induced and venturi draft systems, the only difference being in the type of pressure transformer used. Natural draft systems are self-operating and, unlike induced and venturi systems, require no power to operate in order to produce the pressure difference and maintain the gas flow.

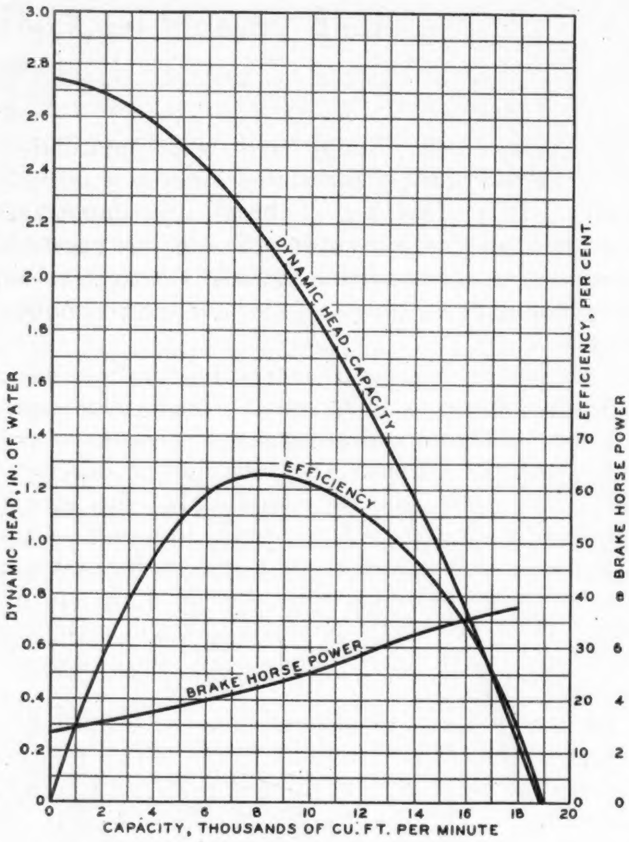

Fig. 1—General operating characteristics of typical induced draft fan.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a public nuisance. The draft as created by a natural draft chimney is a type of draft by aspiration, commonly called suction, as contrasted with forced draft as created by a blower which is a type of draft by blowing. Natural draft is a pressure difference, or head, produced by the difference in weight between the relatively hot gases inside the chimney and a column of atmosphere of equivalent height. The chimney is the pressure transformer and the draft is the result of the thermal difference.

For some unaccountable reason, the performance of a natural draft chimney is generally greatly misunderstood. This is due, no doubt, to the fact that the pressure transformer has not been disassociated from the rest of the plant and considered as a free body. Then, too, it has been the general custom to select the required size of chimney from a table of chimney sizes based on boiler horsepower and, in turn, basing the performance of the chimney on the performance of the boilers, whereas the exact opposite is true, that is to say, the performance of the boilers is based on the performance of the chimney. As a matter of fact, no part of a steam plant has been made the victim of such a lot of nonsensical and meaningless theory as the chimney.

In previous paragraphs, it was stated that a natural draft system produced the same results as an induced draft system; that, in the main, the general theory of draft could be attacked from the standpoint of the theory of hydraulics of liquids; and that, as an agitating agency, a chimney serves the same purpose with gases that a centrifugal pump serves with water. Since a chimney, in general, serves the same purpose and produces the same results as a fan or a centrifugal pump, the question naturally arises as to why the performance of the former is not generally similar to that of the latter. From this inference, the startling but none-the-less true statement can be made that the performance of a natural draft chimney is exactly similar to that of a fan or a centrifugal pump, more particularly the latter.

Figs. 1, 2 and 3 show the general operating characteristics of a typical induced draft fan, a typical centrifugal pump and a typical natural draft chimney, respectively. Each set of curves consists, in general, of a head-capacity curve, an efficiency curve and a horsepower curve.

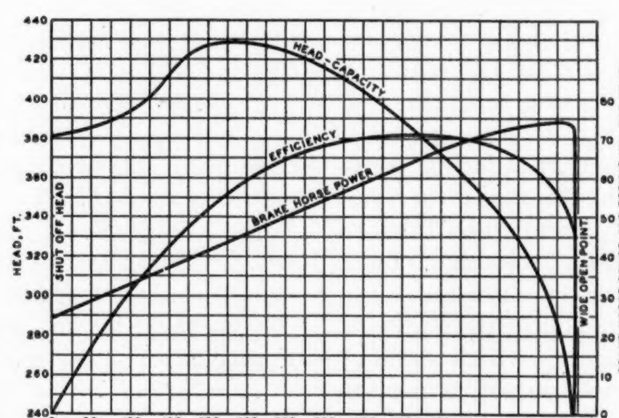

Fig. 2—General operating characteristics of typical centrifugal pump.

The draft-capacity curve of a natural draft chimney corresponds to the head-capacity curve of a centrifugal pump or a fan, draft and head being synonomous. These curves show the variation in the capacity with changes in the head, or draft, and,

vice versa, the variation in the head, or draft, with changes in the capacity.

The efficiency curve of a natural draft chimney corresponds to the efficiency curve of a centrifugal pump or a fan. These curves represent the ratio of the energy output to the energy input.

The theoretical horsepower curve of a natural draft chimney corresponds to the brake horsepower

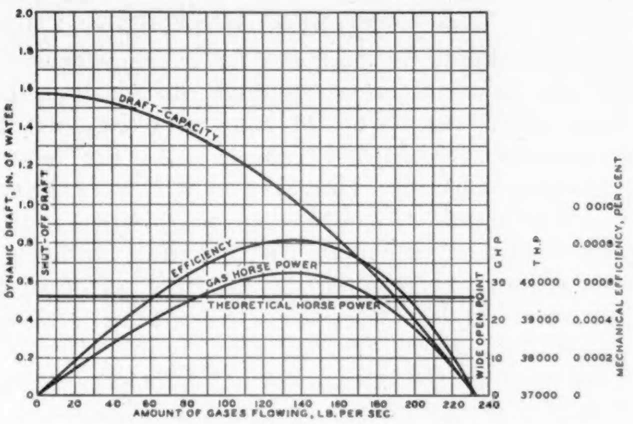

Fig. 3—General operating characteristics of typical natural draft chimney.

curves of a centrifugal pump or a fan. These curves represent the sum total of the energy input supplied by the prime mover.

In view of the analogous factors related above, a natural draft chimney may be regarded as a gas pump operating with the same relative characteristics as those of a centrifugal pump or a fan.

Considering the pump and fan characteristics, the point of zero capacity is called the shut-off head, shut-off pressure, or head of impending delivery, and the point of zero head, the wide open point. With natural draft chimney characteristics, the point of zero capacity is the theoretical draft intensity and is called the shut-off draft or draft of impending delivery. The point of zero draft is called the wide open point.

The head-capacity curve of a centrifugal pump usually rises slightly from shut-off head to a maximum head some distance out after which the curve droops to the wide open point. The maximum head developed by a fan or a centrifugal pump is not necessarily the shut-off head. On the other hand, the draft-capacity curve of a natural draft chimney droops from the maximum at shut-off draft to the wide open point. The maximum head, or draft, developed by a natural draft chimney is always the shut-off, or theoretical, draft.

The performance of a natural draft chimney of a given size expressed in terms of a height and diameter in feet is dependent upon the following factors:

1. Chimney gas temperature,
2. Chimney gas density,
3. Atmospheric temperature,
4. Elevation of plant above sea level,
5. Amount of gases flowing.

The chimney gas temperature (1) depends upon the amount of heat extracted by the boiler and the accessories; the density of the chimney gases (2), upon the character and constituents of the fuel burned; the atmospheric temperature (3), upon the weather conditions; the elevation of the plant above sea level (4), upon the location of the plant; and the amount of gases flowing (5), upon the amount of fuel burned and the excess air supply. The density of the chimney gases will average approximately 0.09 lb. per cu. ft. at 0 deg. fahr. The great majority of the steam plants of the country are built at an approximate sea level elevation or at such a relatively low elevation that the atmospheric pressure will have no appreciable effect upon the chimney performance. Hence, in general, the only variable factors which appreciably affect the performance of a chimney are the chimney gas temperature, the atmospheric temperature and the amount of gases generated and flowing.

Fig. 4 shows the variation in the chimney performance due to changes in the chimney gas temperature with a constant atmospheric temperature. An inspection of these curves discloses the fact that the static draft and also the mechanical efficiency increase without limit as the chimney gas temperature increases. On the other hand, the wide open point and the mechanical efficiency increase from a capacity standpoint as the chimney gas temperature increases until a maximum is reached at a gas temperature of around 584 deg. fahr., after which

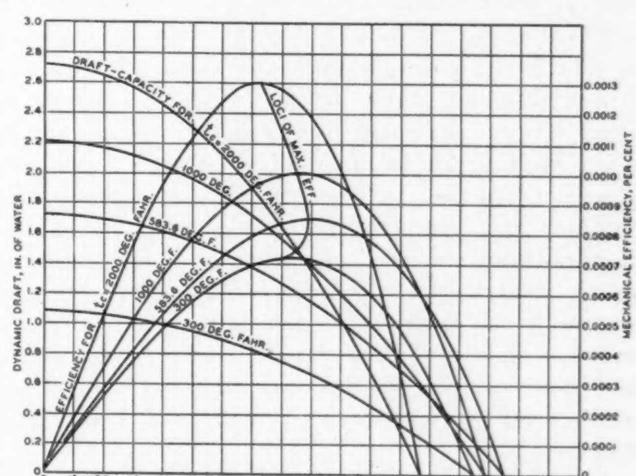

Fig. 4—Variation in chimney performance due to changes in gas temperature with constant atmospheric temperature.

they decrease. Hence, in general, a natural draft chimney is most efficient in performance from a mechanical efficiency standpoint at a chimney gas temperature of around 585 deg. fahr. This figure is based on an atmospheric temperature of 62 deg. fahr. and varies somewhat, as will be seen later.

The variation in the chimney gas temperature of a natural draft chimney produces the same relative results as a variation in the speed of the impeller of a centrifugal pump, or a fan or blower. The head and the capacity of a centrifugal pump, for example,

vary as the speed at which the impeller is driven, the former as the square of the speed and the latter directly as the speed. For any speed, there is a point at which the pump is most efficient and this point should be, and usually is, at the capacity at which the pump normally operates with the proper head desired. At points to the left of this maximum on the efficiency curve, the capacity will be less, the head developed by the pump greater and the corresponding brake horsepower less while to the

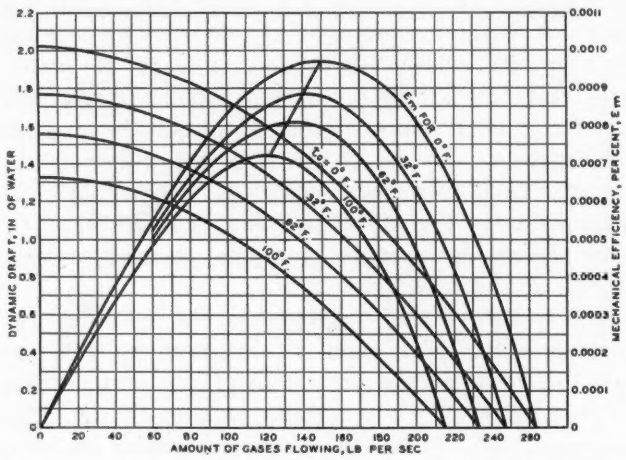

Fig. 5—Variation in chimney performance due to changes in atmospheric temperature with constant gas temperature.

right of this maximum point, the capacity will be greater, the head developed less and the corresponding brake horsepower greater. Likewise, in a natural draft chimney, the point of maximum mechanical efficiency should coincide with the capacity at which the chimney normally operates with the proper dynamic draft desired. At this point, in other words, the ratio of the B.t.u. output to the B.t.u. input will be greatest. To secure the best results, or to get the most out of the chimney, the chimney gas temperature should be kept in the neighborhood of 585 deg. fahr. at a mean atmospheric temperature of 62 deg. fahr. However, the maximum efficiency at 500 deg. fahr. is only slightly less than that at 585 deg. fahr., and the former figure may therefore be considered as the most economical temperature at which to operate from a practical standpoint.

As stated in the previous paragraph, the maximum mechanical efficiency of a chimney should be at the normal working capacity. At points to the left of this maximum, the capacity will be less, the dynamic draft developed greater, and the gas horsepower less, while to the right of this maximum point the capacity will be greater, the dynamic draft developed less, and the gas horsepower greater. This is equivalent to the usual expression of saying that a chimney is overloaded or overworked when it is operating on points to the right of the maximum efficiency point and is operating at "under-capacity" on points to the left.

Fig. 5 shows the variation in the performance of a natural draft chimney due to changes in the atmospheric temperature with a constant chimney gas temperature. These curves show that the static draft, the dynamic draft and the maximum mechanical efficiency all increase as the atmospheric temperature decreases and the chimney gas temperature remains constant. This explains why the draft of a chimney is always better during a cool period or during a sudden temperature drop. From these curves, it would be inferred that a natural draft chimney should be designed for a temperature of 100 deg. fahr. for the atmosphere, inasmuch as this figure represents about the limit and the chimney performance would be poorest, all other factors remaining constant. However, as will be explained in a later article, it would be a rare condition indeed that would impose upon an operating chimney a maximum capacity condition during a period of an extremely high atmospheric temperature and for this reason a figure of 62 deg. fahr., which represents a fair mean yearly temperature in the temperate zone, is used in determining the proper size. When such a figure has been used, then the dynamic draft developed by the chimney, likewise the capacity and the maximum mechanical efficiency, will be greater at temperatures below this mean, and less at temperatures greater. The atmospheric temperature variation in the performance of a natural draft chimney corresponds roughly to the "over-capacity" condition of a centrifugal pump. In both cases, the wide open point is extended further to the right as the atmospheric temperature decreases in the one instance and as certain mechanical limitations are provided in the other.

The variations in the performance of a natural draft chimney due to changes in the chimney gas density, the assumed value for the coefficient of friction, and the atmospheric pressure for relatively low elevations are, for the most part, of slight degree, and, generally speaking, are compensating.

Very few operators would operate a centrifugal pump, or a fan for that matter, without a set of performance curves, or operating characteristics, of the machine at hand. The actual performance can be checked instantly from these characteristics which were developed on test. Likewise every operating engineer should have a set of operating characteristics of his natural draft chimney so that the actual operation of the chimney can be likewise checked instantly. The actual dynamic draft which the chimney develops is of course indicated by the draft gage. This is equivalent to the head which is developed by a centrifugal pump and which is indicated by a pressure gage. In neither case, however, can the actual capacity of the pressure transformer be determined without a set of the operating characteristics.

After the values for the chimney gas temperature, the atmospheric pressure and the atmospheric tem-

(*Continued on page 39*)

Direct Firing Steel Plant Boilers with Roller Mills

By
HENRY KREISINGER*
and
A. C. FOSTER*

The direct-fired method of burning pulverized fuel has achieved marked popularity. The evolution of this method in the comparatively few years of its existence has been exceedingly rapid, and there is no question but that remarkable progress has been made in improving the design and performance of the equipment comprising such systems. Various types of pulverizing mills have been used with varying degrees of success and certain mills have become recognized as being well suited to the requirements of direct-firing. Until recently the roller mill, long regarded as ideally suited to storage system applications, has not been included in this group. Within the past year, however, several notably successful installations have been made. One of these installations is described in the accompanying article. In this plant a direct-fired system with roller mills and automatic combustion control has proved itself to be highly practicable, reliable and efficient.

THE Pittsburgh Steel Company has recently placed in operation a new steam plant at Monessen, Pennsylvania. The plant consists of four three-drum Ladd bent tube boilers of 14,250 square feet of heating surface each. The novel feature of this installation is that the direct-fired method of burning pulverized fuel is used in combination with roller mills and automatic combustion control. There are two four-roller mills to each boiler, each mill supplying pulverized coal to one horizontal forced-draft round burner. Boilers No. 3 and No. 4 in addition to the coal burners are equipped with Steinbart burners for burning blast furnace gas. These burners are located in the front wall below the coal burners. The general arrangement of the various equipment comprising these two units is shown in Figs. 1 and 2. Boilers No. 1 and No. 2 are equipped for burning pulverized coal only.

The furnaces are of the hollow air-cooled wall design with ash hoppers at the bottom. The air passing through the hollow walls is heated to about 140 deg. fahr., and is used for drying the coal in the mills. It is then blown with the pulverized coal into the furnaces as primary air. The ash collected in the ash hoppers of the furnaces are removed by a sluicing system.

The mills are of the standard Raymond roller mill design, with four rollers automatically lubricated with oil, and having a maximum capacity of four tons per hour of Pittsburgh coal. The mills are located in the basement of the boiler plant, but the mill blowers and mill feeders are placed on the operating floor which is nearly on the level with the coal burners as shown in Fig. 1. On the operating floor are also placed 8 ten-gallon oil tanks, one for each mill, which serve as reservoirs for the lubricating oil for the mills. The oil flows by gravity to the mills while they are in operation, and the mills can remain in operation indefinitely as far as lubrication is concerned.

All combustion control apparatus and instrument panels are located on the operating floor, within easy access of the operators.

*Combustion Engineering Corporation, New York.

The boilers are operated under the widely fluctuating load condition typical of steel plants. The variation in the demand for steam is met entirely by the regulation of the coal pulverizing and burning equipment. Boilers No. 3 and No. 4 burn all the blast furnace gas that is available for this plant without regulation with relation to the steam demand on the plant. The pulverized coal equipment is regulated automatically with such nicety that practically constant steam pressure is maintained. Charts 1 to 6 of Fig. 3 show how well the pulverized coal equipment is controlled. Charts 1 to 4 give the steam flow from each of the four boilers. Chart 5 shows the flow of steam from the plant and available for use in the steel mill. This chart does not include steam used in auxiliaries in the steam plant. Attention is called to the flat tops of some of the peaks of Chart 5. The steam flow on these peaks exceeded the capacity of the steam flow chart, otherwise the peaks would have been shown higher and more prominent. Chart 6 shows the steam pressure in the main at the steam plant. The variation in pressure does not exceed 10 lb.

Increased demand for steam is met by an increased rate of milling and delivery of pulverized coal to the boiler furnaces. The increased rate of milling is obtained by speeding up the mill and the mill blower. With the Raymond roller mill the rate of pulverization varies approximately as the cube of the speed, so that small variation in the speed gives a comparatively large variation in the output of the mill without materially changing the fineness of the coal.

Generally a variation in speed from 70 to 100 per cent is accompanied with a variation in the mill output from 33 to 100 per cent.

The control system used is of Hagan manufacture and the sequence of operation is as follows:

Greater demand for steam causes the pressure in the main steam header to drop below normal which through the master control increases the speeds of the mills, mill blowers, and mill feeders, and admits more secondary air, to the burners by opening wider the dampers at the burners. The induced draft fan control tends to maintain a constant furnace draft. As the furnace draft decreases with the addition of more fuel and air, the induced draft fan speeds up simultaneously with the opening of the secondary air dampers. This process continues until normal steam pressure and normal furnace draft are re-established. There is practically no time lag in the operation of the induced draft fan control. A change in furnace draft of 0.01 in. of water is sufficient to cause a speed change in the induced draft fan.

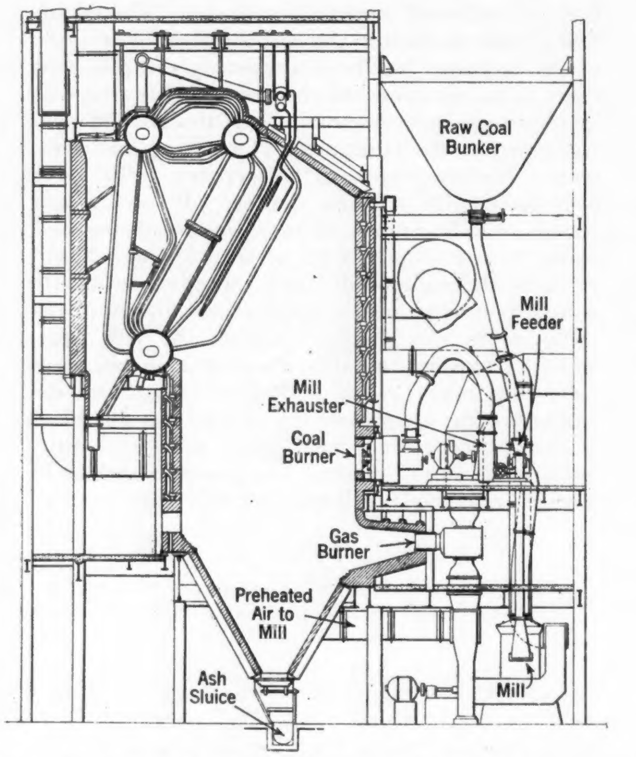

Fig. 1—Section through plant showing the general arrangement of the equipment forming one of the steam generating units burning blast furnace gas and coal.

In case the demand for steam decreases, the steam pressure in the main header rises, and through the master control, slows down the mills, mill blowers, and mill feeders, and reduces the secondary air by partly closing the damper at the burners. The resultant tendency of the furnace draft to increase causes the induced draft fan to be slowed down. These actions continue until normal pressure is re-established. All these actions take place with a variation in pressure not exceeding 10 lb. as shown by Chart 6 of Fig. 3.

The nicety with which the combustion control operates is indicated by the percentage of CO_2 in the products of combustion in the first pass of the boiler and the furnace drafts given in the following Table.

Steam flow—lb. per hr.	Per cent CO_2 in gases with coal firing only	Draft in furnace—in. of water
46,000	13.7	0.05
80,000	13.0	0.15
120,000	12.3	0.20

The motors driving the mills are d.c. shunt-wound variable speed from 300 to 500 r.p.m. and 45 to 60 hp. There are 39 points on the rheostat control. The usual operating range is from point 7 to point 23, and in speed from 337 to 425 r.p.m.

The speed of the mill blowers is varied through about the same range as the speed of the mills. The blowers are driven by d.c. shunt-wound variable speed motors with speed range of 950 to 1250 r.p.m. and 25 to 35 hp. There are 39 points on the rheostat, the usual operating range being from point 8 to point 39.

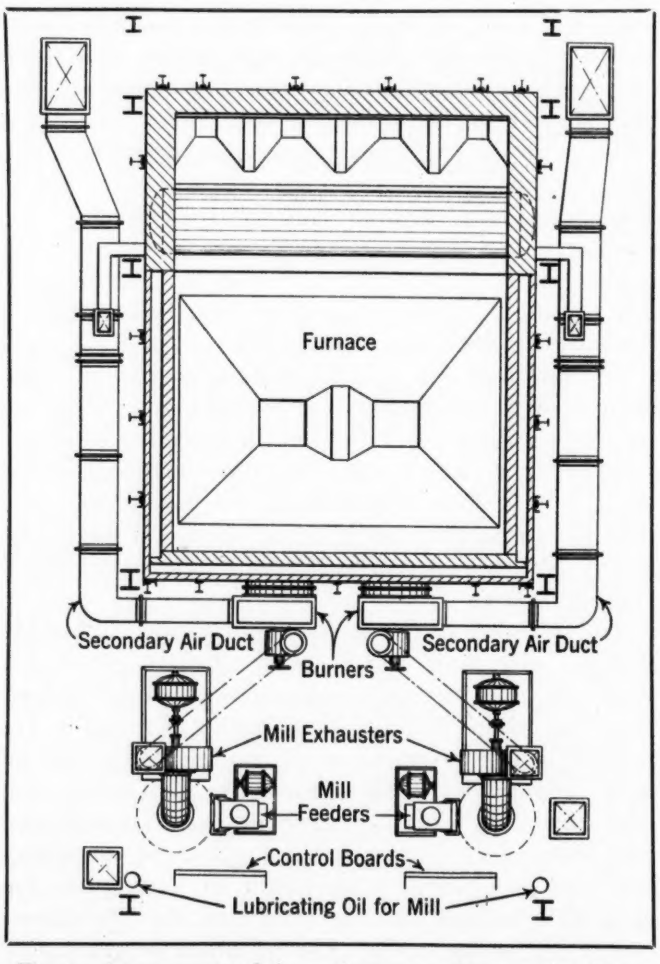

Fig. 2—Arrangement of the equipment on the operating floor.

Coal is fed to the mills approximately at the same rate as the mill pulverizes it. The rate of feeding is varied by the variation of the speed of the motor driving the feeder. The motor driving the feeder is a d.c. shunt-wound variable speed motor with speed from 500 to 2000 r.p.m. and ¾ to 2 hp. There

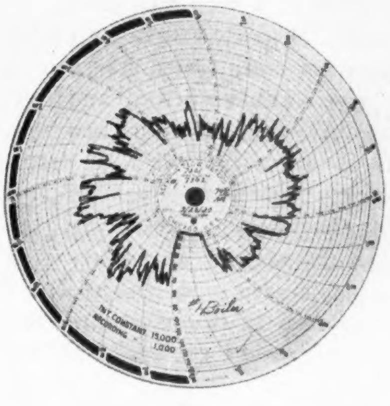

CHART No. 1— Steam flow from Boiler No. 1.

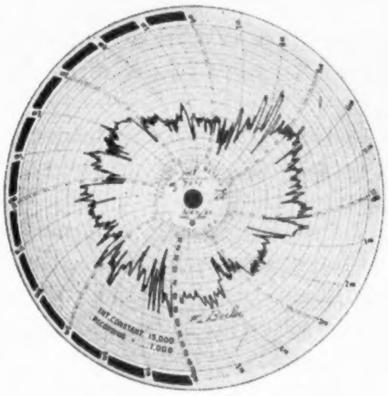

CHART No. 2— Steam flow from Boiler No. 2.

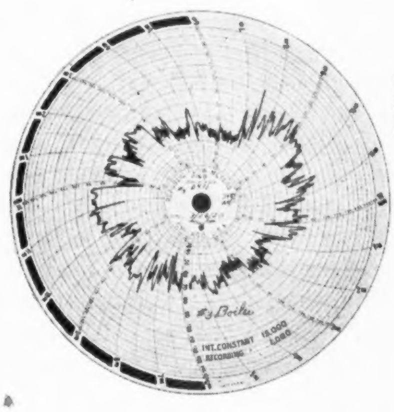

CHART No. 3— Steam flow from Boiler No. 3.

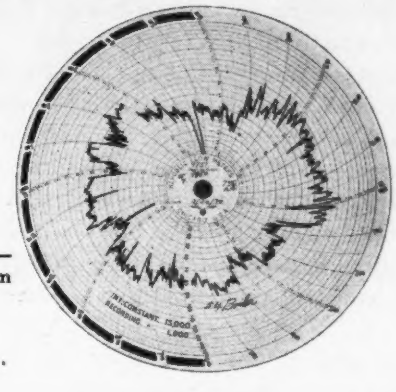

CHART No. 4— Steam flow from Boiler No. 4.

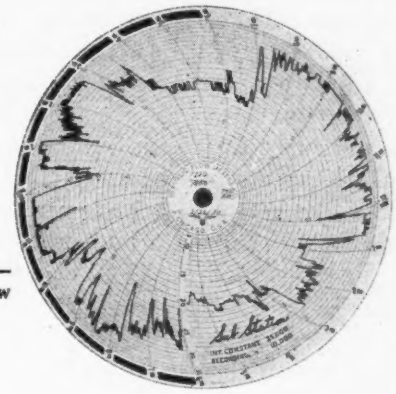

CHART No. 5— Total steam flow from plant.

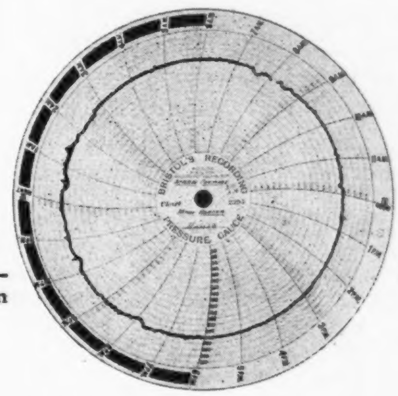

CHART No. 6— Pressure in main steam line.

Fig. 3—Charts showing the load on the steam generating units, and the precision with which the combustion control operates. In spite of the greatly varying load the pressure remains practically constant during a 24 hour run.

are 39 points on the rheostat. The usual operating range is from point 1 to point 19, and from 635 to 1175 r.p.m.

All motors are designed to operate on 230 volt current.

Besides the variable speed control on the mill feeder, there is the usual feed control which automatically cuts out the feeder, in case the mill becomes too full of coal, and cuts it in again when the amount of coal in the mill has been reduced to normal. This control is operated by the air pressure drop through the mill. If the mill becomes too full of coal the pressure drop through the mill is increased to a point where it shuts down the feeder motor. The motor is again started when the pressure drop through the mill becomes normal.

The air passing through the mill is the primary air used for combustion, and therefore the amount of primary air is regulated by the mill blower control. This pressure of the primary air is from 3.5 to 7 in. of water at the burner.

Secondary air for combustion is supplied at a pressure of about 6.5 in. of water by one fan per boiler, each fan having a capacity of 55,400 c.f.m. and driven by a steam turbine. These turbines are designed to operate on either 375 lb. pressure and

200 deg. fahr. superheat, or 150 lb. pressure saturated steam, with an exhaust pressure of 6 lb. in each case. The speed of the fan is varied from 936 to 1672 r.p.m., which speeds require 15 to 90 hp. The fans are of double-inlet full-housing design with shrouded wheel and curved blades. A constant pressure in the secondary air duct is maintained by Hagan automatic control with a double regulation, one controlling the damper in the fan discharge and the other the turbine speed. The controls are so adjusted that the damper is nearly fully open most of the time so as to avoid unnecessary resistance to the flow of air.

The amount of secondary air supplied to the burners is automatically controlled by a damper at each burner. The pressure in the wind box of the burner varies from 1.5 to 6.5 in. of water. No automatic adjustment of the burner proper is needed. The burner is adjusted by hand for the average rating and this adjustment is not changed during the normal operating conditions.

Each boiler unit is provided with one induced draft fan driven by a steam turbine through a reduction gear. The fans on units Nos. 1 and 2, burning only pulverized coal, have a capacity of 141,200 c.f.m. at 4.0 in. pressure and 670 deg. fahr. temperature. The fans on units Nos. 3 and 4 burning blast furnace gas and pulverized coal have a capacity of 240,200 c.f.m. at 5.0 in. pressure and 900 deg. fahr. temperature. All the induced draft fans are of the double inlet and straight blade wheel type, which is the type least eroded by the fly ash.

The speed variation of the induced draft fans is shown in the following Table:

Boilers 1 and 2—Pulverized coal only

Rating per cent	hp.	Turbine r.p.m.	Fan r.p.m.
—	157*	3117*	322*
300	81	2555	264
250	37	2004	207
200	20	1636	169
150	10	1226	137

Boilers 3 and 4
Blast furnace gas and pulverized coal

Rating per cent	hp.	Turbine r.p.m.	Fan r.p.m.
—	217*	3562*	368*
210	127	2982	308
200	108	2875	297
150	46	2381	246

*Max. for design

The capacity of units 3 and 4, when operating on blast furnace gas alone, is limited by the temperature of the products of combustion leaving the boiler, which at 210 per cent of rating is about 800 deg. fahr. This is the highest temperature at which the induced fan wheels, which are made of ordinary steel, are safe to operate. For higher ratings, with blast furnace gas alone and high temperature of flue gases, wheels of special alloy would be required.

The induced and forced draft fans are larger than at present needed, provision having been made for the installation of air heaters at some future date.

The roller type of mill is well adapted for direct-firing pulverized coal. It gives a uniform fineness of coal under varying load conditions, has low maintenance and low power consumption, and operates quietly. With the automatic oil lubrication, it can be kept in continuous operation over long periods of time. It is particularly well suited for direct-firing wet coals. During the normal operation there is a bed of coal about 4 in. high at the bottom of the mill, which serves as a reservoir for the mill to draw on in case the feed of coal to the mill is interrupted by the clogging of the feeder, or wet coal sticking in the spout delivering coal from the raw coal bin to the mill feeder. If similar interruption in feed occurs in the high speed type of mill, which operates with practically no coal reserve in the mill, the fire may go out.

Interruption and irregularity in the feed of coal to the mill is particularly apt to occur at low rates of feeding and milling. The coal is moving slowly through the spouts and the feeder, and may stop alogether. Such interruptions in the coal supply to the mill are much less serious with the roller type of mill than they are with other types of mill. The roller mill holding a coal reserve keeps on milling coal at a rate corresponding to its speed, and is able to supply the furnace with coal at the desired rate for three or four minutes, during which time the cause of interruption in the coal feed can be removed. At higher rates of milling, the coal reserve in the mill does not last as long as at low ratings, but lasts sufficiently long to span most of the ordinary interruptions in the coal supply to the mill.

In the roller type of mill, the rate of milling depends almost entirely on the speed of the mill, and is to a large extent independent of the rate of feeding the coal to the mill. If the coal is fed faster than the mill can pulverize it, the mill eventually becomes too full of coal and the automatic feed regulator stops the feeder until the amount of coal in the mill is reduced to normal. In case the coal is fed to the mill at slower rate than the mill is pulverizing it, due to an obstruction in the feeder or coal spout, the mill draws on its coal reserve and gradually becomes noisy as it works out its coal reserve, letting the operator know that something is wrong with the coal supply to the mill. Thus, a steady and even output is maintained from the mill, even with very irregular feed to the mill that is likely to occur with wet coal, or a greatly varying size of coal.

This coal reserve in the roller type of mill has also an advantage for mill drying. The coal stays in the mill one or two minutes, during which time it is constantly agitated by the mill plows and exposed to the drying action of the air. Consequently a more thorough drying of coal can be accomplished than with any other type of mill.

Thermal Changes in Gases*

By Wm. L. DE BAUFRE, International Combustion Engineering Corporation, New York

This is the eighth in the series of articles by Mr. DeBaufre and the third on the subject of gases. The present article deals with thermal changes in gases where flow of the gas is not involved. The next article will be devoted to heat engines and the second law of thermodynamics. In future articles, the frictional resistance to the flow of fluids and thermal changes involved in such flow will be discussed.

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THE various mathematical formulas used by engineers to calculate thermal changes in gases are in general based on the pressure-volume-temperature relation of a perfect gas and on the assumption of constant specific heats. For low and moderate pressures and for small temperature changes, these formulas are sufficiently accurate for practical purposes. For large changes in temperature, however, the accuracy now demanded in design and performance calculations does not permit the assumption of constant specific heats. Variable specific heats as functions of temperature may be combined with the pressure-volume-temperature relation of a perfect gas for more nearly accurate calculations in all except a few industrial processes where very high pressures are employed. In these few cases, the effects of very high pressures upon both the specific heats and the volumes and temperatures of gases must be taken into account, preferably by the use of tables or curves giving the thermal properties of the gases employed, such as those in use for steam. If such tables or curves are not available, the desired effects of very high pressures can be approximated by methods touched upon in this article which will, however, be confined mainly to a discussion of thermal changes based on the pressure-volume-temperature relation of a perfect gas and either on constant specific heats or on specific heats which vary with temperature only.

Thermal Changes under Constant Pressure and at Constant Volume

The heating or cooling of gases under pressures which vary so slightly that they may be considered constant from the standpoint of addition or abstraction of heat, is probably the most common thermal change occurring in gases in industrial processes. For monatomic gases, such as mercury, argon and helium, and for other gases when the use of constant specific heats gives sufficient accuracy for the work in hand, the calculation of the heat added to or abstracted from a gas under constant pressure is a very simple matter as indicated by the mathematical expression

$$Q = C_p \times W \times (t_2 - t_1)$$

where t_1 = initial temperature of the gas,
t_2 = final temperature of the gas,
W = weight of gas heated,
C_p = specific heat of unit weight of the gas under constant pressure and
Q = heat absorbed by the gas.

Thus, to calculate the heat discharged from a steam boiler in the products of combustion passing to the stack, the A.S.M.E. Boiler Test Code gives a constant specific heat of 0.24 B.t.u. per lb. per deg. fahr. for the dry constituents and a constant specific heat of 0.46 B.t.u. per lb. per deg. fahr. for the water vapor therein. In burning a certain Pittsburgh bituminous coal, 13.85 lb. of products of combustion were produced, including 0.60 lb. of water vapor. The coal had a heating value of 13700 B.t.u. per lb. In order to increase the boiler efficiency one per cent, it would be necessary to cool the products of combustion before they are discharged to the stack an additional number of degrees $(t_2 - t_1)$ as given by $0.01 \times 13700 = 0.24 \times 13.25 \times (t_2 - t_1) + 0.46 \times 0.60 \times (t_2 - t_1)$; whence, $(t_2 - t_1) = 39.6$ deg. fahr.

While it is permissible to use a constant specific heat to calculate the loss in boiler efficiency due to the heat discharged in the stack gases which rarely escape at temperatures above 500 deg. fahr., the accuracy required in design calculations for boilers where temperatures of the products of combustion up to 2500 deg. fahr. or more must be considered, necessitate taking into account the increase in the specific heats of gases with rise in temperature. For the most common gases encountered in combustion calculations, the following formulas by Goodenough and Felbeck may be used for constant pressure heating or cooling, as given in the previous article on the thermal properties of gases, namely:

For carbon dioxide,

$$C_p = 6.548 + 5.067\left(\frac{T}{1000}\right) - 1.248\left(\frac{T}{1000}\right)^2 + 0.1085\left(\frac{T}{1000}\right)^3$$

For water vapor,

$$C_p = 8.33 - 0.276\left(\frac{T}{1000}\right) + 0.423\left(\frac{T}{1000}\right)^2$$

For the diatomic gases, nitrogen, oxygen and carbon monoxide.

$$C_p = 6.93 + 0.1200 \left(\frac{T}{1000}\right)^2$$

In the above formulas, T is the temperature in degrees fahr. absolute and C_p is the specific heat per lb.-mole in B.t.u. per deg. fahr.

To find the heat added or abstracted under constant pressure between any two temperatures T_0 and T, the appropriate relation for C_p is substituted in $dQ = C_p\, dT$ and the expression integrated between the limits T_0 and T, giving: For carbon dioxide,

$$\begin{aligned} Q = 6548 & \left[\left(\frac{T}{1000}\right) - \left(\frac{T_0}{1000}\right)\right] \\ + 2533.5 & \left[\left(\frac{T}{1000}\right)^2 - \left(\frac{T_0}{1000}\right)^2\right] \\ - 416 & \left[\left(\frac{T}{1000}\right)^3 - \left(\frac{T_0}{1000}\right)^3\right] \\ + 27.125 & \left[\left(\frac{T}{1000}\right)^4 - \left(\frac{T_0}{1000}\right)^4\right] \end{aligned}$$

For water vapor,

$$\begin{aligned} Q = 8330 & \left[\left(\frac{T}{1000}\right) - \left(\frac{T_0}{1000}\right)\right] \\ - 138 & \left[\left(\frac{T}{1000}\right)^2 - \left(\frac{T_0}{1000}\right)^2\right] \\ + 141 & \left[\left(\frac{T}{1000}\right)^3 - \left(\frac{T_0}{1000}\right)^3\right] \end{aligned}$$

For the diatomic gases, nitrogen, oxygen and carbon monoxide,

$$\begin{aligned} Q = 6930 & \left[\left(\frac{T}{1000}\right) - \left(\frac{T_0}{1000}\right)\right] \\ + 40 & \left[\left(\frac{T}{1000}\right)^3 - \left(\frac{T_0}{1000}\right)^3\right] \end{aligned}$$

The values in Table I were calculated by the above formulas for T_0 = 459.6 fahr. absolute and represent the heats added above zero fahrenheit to the respective gases under constant pressure. Values for 32 fahr., 60 fahr. and 70 fahr. are included in the table in order that the heat added above any one of these reference temperatures may be readily obtained by subtracting one of these values from those tabulated for higher temperatures. To obtain the heat added per pound of gas, divide the tabulated values by 44 for carbon dioxide, by 18 for water vapor, by 28 for nitrogen, by 32 for oxygen and by 28 for carbon monoxide. To obtain the heat added per pound of dry atmospheric air, divide the heat added to the diatomic gases by 28.97, the equivalent molecular weight of dry atmospheric air. The heat added to a cubic foot of any of the gases may be found by dividing the tabulated values by the volume of one lb.-mole of gas, namely, 359.0 cu. ft. at 32 fahr. under one atmosphere pressure or 379.5 cu. ft. at 60 fahr. and one atmosphere pressure.

By means of the values in Table I, the sensible heats were calculated for several temperatures of the

TABLE I. SENSIBLE HEATS OF GASES

Temperature, fahr.	Heat added under constant pressure, B.t.u. per lb.-mole			External work under constant pressure, B.t.u. per lb.-mole
	CO_2	H_2O	N_2O_2, CO (dry air)	
0	0.0	0.0	0.0	0.0
32	278.0	265.4	222.6	63.5
60	524.5	497.8	417.5	119.1
70	613.3	580.8	487.2	139.0
100	882.0	830.0	696.1	198.5
200	1,801.6	1,661.9	1,393.6	397.0
300	2,756.9	2,496.6	2,092.6	595.5
400	3,745.8	3,335.1	2,793.5	794.0
500	4,766.4	4,178.0	3,496.5	992.5
600	5,816.6	5,026.3	4,201.7	1,191.0
700	6,894.7	5,880.8	4,909.5	1,389.5
800	7,999.0	6,742.3	5,620.1	1,588.0
900	9,127.6	7,611.8	6,333.6	1,786.5
1000	10,279.0	8,489.9	7,050.5	1,985.0
1100	11,451.6	9,377.7	7,770.9	2,183.5
1200	12,643.8	10,275.9	8,495.0	2,382.0
1300	13,854.3	11,185.4	9,223.0	2,580.5
1400	15,081.6	12,107.0	9,955.3	2,779.0
1500	16,324.4	13,041.6	10,692.1	2,977.5
1600	17,581.4	13,990.0	11,433.6	3,176.0
1700	18,851.6	14,953.0	12,180.0	3,374.5
1800	20,133.7	15,931.6	12,931.6	3,573.0
1900	21,426.6	16,926.5	13,688.6	3,771.5
2000	22,729.5	17,938.7	14,451.3	3,970.0
2100	24,041.4	18,968.8	15,219.9	4,168.5
2200	25,361.4	20,017.9	15,994.6	4,367.0
2300	26,688.7	21,086.7	16,775.7	4,565.5
2400	28,022.6	22,176.1	17,563.5	4,764.0
2500	29,362.4	23,287.0	18,358.1	4,962.5
2600	30,707.5	24,420.1	19,159.8	5,161.0
2700	32,057.4	25,576.3	19,968.8	5,359.5
2800	33,411.6	26,756.5	20,785.4	5,558.0
2900	34,769.7	27,961.5	21,609.9	5,756.5
3000	36,131.3	29,192.2	22,442.4	5,955.0
3100	37,496.3	30,449.4	23,283.2	6,153.5
3200	38,864.3	31,733.9	24,132.6	6,352.0
3300	40,235.2	33,046.7	24,990.7	6,550.5
3400	41,609.0	34,388.5	25,857.9	6,749.0
3500	42,985.6	35,760.2	26,734.3	6,947.5
3600	44,365.1	37,162.6	27,620.3	7,146.0
3700	45,747.6	38,596.6	28,515.9	7,344.5
3800	47,133.2	40,063.0	29,421.6	7,543.0
3900	48,522.3	41,562.8	30,337.5	7,741.5
4000	49,915.1	43,096.6	31,263.8	7,940.0

TABLE II. ACTUAL AND APPARENT SENSIBLE HEATS IN PRODUCTS OF COMBUSTION

Temperature of products of combustion, fahr.	Actual sensible heat above 70 fahr., B.t.u. per lb. of coal	Apparent sensible heat calculated by constant specific heats, B.t.u. per lb. of coal
100	103	104
200	447	449
300	794	795
400	1144	1140
500	1497	1486
600	1853	1832
1000	3304	3214
1500	5183	4942
2000	7141	6670
2500	9182	8398
3000	11316	10126

13.85 lb. of products of combustion of the Pittsburgh coal previously mentioned. These have been tabulated together with the apparent sensible heats calculated by means of the constant specific heats given in the A.S.M.E. Boiler Test Code. The error in boiler efficiency based on the latter values is evidently small up to 500 or 600 fahr. stack temperature. For products of combustion leaving the furnace at say 2000 fahr., however, the difference between the actual and apparent sensible heats is too great to use the latter in design calculations.

If, instead of permitting the volume of a gas to increase with rise in temperature, the gas be confined within a vessel so that it cannot expand and perform external work while being heated, the heat added between any two temperatures will be less than that added under constant pressure by the quantity

$$(C_p - C_v) \times W \times (t_2 - t_1)$$

where t_1 = initial temperature of the gas,
t_2 = final temperature of the gas,
W = weight of gas heated,
C_v = specific heat at constant volume and
C_p = specific heat under constant pressure.

As explained in a previous article, the difference between the two specific heats $C_p - C_v$ may be taken equal to 1.985 B.t.u. per lb.-mole per deg. fahr. for carbon dioxide and water vapor under low pressures and for air and the diatomic gases under low and even moderately high pressures. We can therefore obtain the heat added to these gases at constant volume by subtracting $1.985\,(t_2 - t_1)$ B.t.u. per lb.-mole from the heat added per lb.-mole under constant pressure.

To facilitate the calculation of heat added under constant volume to the gases listed in Table I, values of 1.985 times the temperature change above zero fahrenheit have been added to the table. These values also represent the external work done under constant pressure heating because these gases may be considered as conforming with the perfect gas pressure-volume-temperature relation under low and even under moderate pressures. Thus, for the initial conditions, $PV_1 = A\,R\,T_1$, and for the final conditions, $PV_2 = A\,R\,T_2$; consequently, the external work done under constant pressure P while heating the gas from T_1 to T_2 is given by $P\,(V_2 - V_1) = A\,R\,(T_2 - T_1) = 1.985\,(t_2 - t_1)$ B.t.u. per lb.-mole.

Since $C_p - C_v$ is approximately equal to 1.985 B.t.u. per lb.-mole per deg. fahr. for all gases, the mathematical expression for the specific heat C_v at constant volume in terms of the absolute temperature T may be obtained by subtracting 1.985 from the corresponding expression for the specific heat under constant pressure C_p as given above; that is: For carbon dioxide,

$$C_v = 4.563 + 5.067\left(\frac{T}{1000}\right) - 1.248\left(\frac{T}{1000}\right)^2 + 0.1085\left(\frac{T}{1000}\right)^3$$

For water vapor,

$$C_v = 6.345 - 0.276\left(\frac{T}{1000}\right) + 0.423\left(\frac{T}{1000}\right)^2$$

For the diatomic gases, nitrogen, oxygen and carbon monoxide,

$$C_v = 4.945 + 0.1200\left(\frac{T}{1000}\right)^2$$

where T is in degrees fahr. absolute and C_v is in B.t.u. per lb.-mole per deg. fahr.

The heat added to or abstracted from a gas under a constant pressure which is very high or at a constant volume while the gas is under very high pressures, may be calculated with greater accuracy than the preceding formulas would give by using the fundamental relation that the heat added is equal to the change in internal energy plus the external work done. The latter quantity can be calculated from the change in volume and the pressure. The former quantity may be determined by methods which will be explained in the next section of this article.

Internal Energy Change

While the heat added to a gas in order to raise its temperature from some initial value to some final value depends upon whether the gas is heated under constant pressure or at constant volume, the internal energy change is determined by the initial and final conditions only. In fact, the change in internal energy can be calculated from the initial and final pressures, volumes and temperatures of a gas no matter what the character of the intermediate thermal change may have been.

For a perfect gas, the internal energy does not vary with volume (or pressure) at constant temperature. Hence, the internal energy of a perfect gas is a function of temperature only and the change in internal energy can be calculated from the initial and final temperatures. For a perfect gas with constant specific heats, the relation of the internal energy u to the absolute temperature T was shown in a previous article to be

$$u = c_v T = \frac{ART}{k-1} = \frac{Apv}{k-1}$$

where c_v is the specific heat at constant volume, k is the ratio of the specific heats under constant pressure and at constant volume (c_p / c_v), R is the gas constant, A is the reciprocal of the mechanical equivalent of heat and p and v are corresponding values of pressure and volume at absolute temperature T. In changing from some initial temperature T_1 to a final temperature T_2, the change in internal energy with constant specific heats is $\Delta u = c_v(T_2 - T_1) =$

$$\frac{AR}{k-1}(T_2 - T_1) = \frac{A}{k-1}(p_2 v_2 - p_1 v_1)$$

For any gas considered as a perfect gas, AR = 1.985 B.t.u. per lb.-mole per deg. fahr. For air considered as a perfect gas, k may be taken as 1.40. The change in internal energy of air considered as a perfect gas

is therefore 1.985 / 0.4 = 4.96 B.t.u. per lb.-mole per deg. fahr. Since the equivalent molecular weight of dry atmospheric air is 28.97 lb., the internal energy change is 0.171 B.t.u. per lb. per deg. fahr.

Instead of calculating the internal energy change of unit weight of gas, such as one pound or one pound-mole, the internal energy change of the total volume of gas under consideration may be calculated by means of the expression

$$\Delta U = \frac{P_2V_2 - P_1V_1}{J\,(k-1)}$$

where $J = 1/A$. For air we have

$$\Delta U = \frac{P_2\,V_2 - P_1\,V_1}{778.6\,(1.40 - 1)}$$

where P_1 = initial pressure in lb. per sq. ft.,
V_1 = initial volume in cu. ft.
P_2 = final pressure in lb. per sq. ft.
V_2 = final volume in cu. ft. and
ΔU = internal energy change in B.t.u.

For any considerable range in temperature of real gases, the change in internal energy per degree change in temperature cannot be taken as constant except for approximate calculations. A closer approach to the true rate of internal energy change is obtained by assuming it equal to the specific heat at constant volume expressed as a function of temperature but independent of pressure. The total change in internal energy is then equal to

$$\Delta u = C_v\,(T_2 - T_1)$$

where C_v represents some mathematical expression in terms of temperature as given above for the more common gases. The difference in the two specific heats may be taken as constant and equal to 1.985 B.t.u. per lb.-mole per deg. fahr. or it may be calculated from an equation of state as explained in a previous article. Using Berthelot's equation of state, the curves in Fig. 1 for the difference and the ratio of the specific heats have been calculated for dry atmospheric air. The accuracy of the calculated curves is indicated by the experimentally determined points for zero centigrade from the work of Koch.

Still more nearly accurate calculations of the change in internal energy of a gas under very high pressures may be made by considering the expression for the specific heat at constant volume as representing the rate of internal energy change under zero pressure and utilizing an equation of state to determine the change in internal energy between zero pressure and the actual pressures involved. For Berthelot's equation of state, the Gay Lussac effect, as shown by a previous article, is

$$\left(\frac{\partial u}{\partial v}\right)_T = \frac{2\,A\,a}{T\,v^2}$$

Integrating between the volumes v_0 and v, we get for the change in internal energy at constant temperature T,

$$(\Delta u)_T = \frac{4\,A\,a}{T}\left(\frac{1}{v_0^3} - \frac{1}{v^3}\right)$$

For an initial pressure of zero, v_0 = infinity; therefore, the change in internal energy at constant temperature from zero pressure to a pressure corresponding to volume v is, in accordance with Berthelot's equation of state,

$$(\Delta u)_T = -\frac{4\,A\,a}{T\,v^3}$$

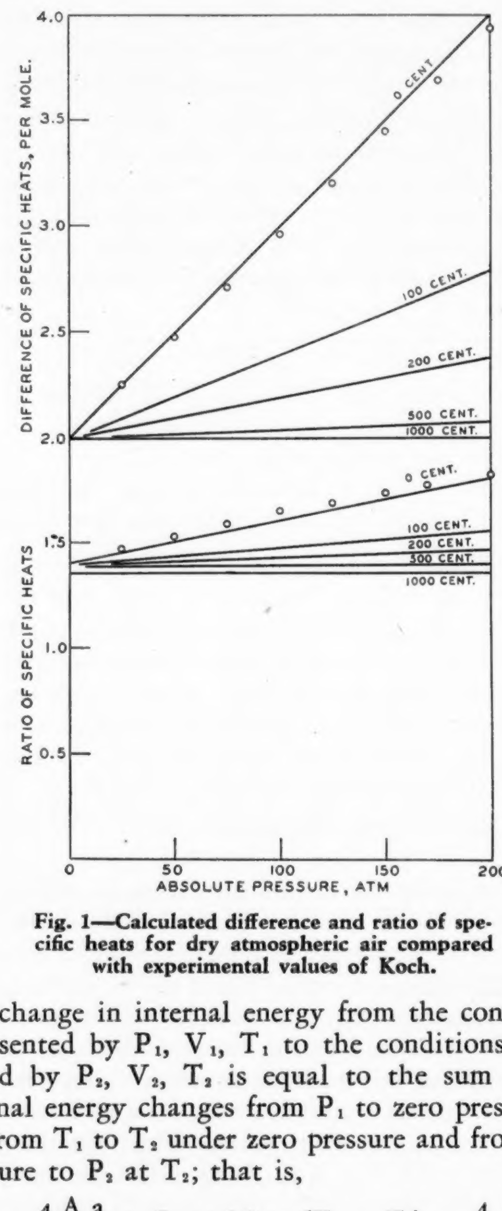


Fig. 1—Calculated difference and ratio of specific heats for dry atmospheric air compared with experimental values of Koch.

The change in internal energy from the conditions represented by P_1, V_1, T_1 to the conditions represented by P_2, V_2, T_2 is equal to the sum of the internal energy changes from P_1 to zero pressure at T_1, from T_1 to T_2 under zero pressure and from zero pressure to P_2 at T_2; that is,

$$\Delta U = \frac{4\,A\,a}{T_1\,V_1^3} + C_v \times M \times (T_2 - T_1) - \frac{4\,A\,a}{T_2\,V_2^3}$$

where M is the number of moles of gas involved in the change.

Isothermal Change

Change in volume with corresponding change in pressure of a gas while the temperature remains constant is known as isothermal expansion or compression. There is probably no industrial process in which isothermal change is closely approached. Isothermal expansion and compression are of great importance, however, by reason of practical deductions which have been based on theoretical discussions involving isothermal changes. Isothermal

changes also serve to represent ideal conditions with which practical performance may be compared.

For a perfect gas, we have the simple relation between pressure and volume during isothermal change represented by Boyle's law, namely,

$$PV = \text{Constant}$$

The value of the "Constant" for any real gas considered as a perfect gas is M R T, where T is the absolute temperature of the gas, R is the gas constant per unit weight and M is the weight (mass) of gas involved in the isothermal change. If the weight M be expressed in lb.-moles and the temperature T in deg. fahr. absolute, then for all gases R = 1545.5 ft. lb. per lb.-mole per deg. fahr. when P and V are expressed in lb. per sq. ft. and cu. ft. respectively. For different temperatures, the isothermal lines of a perfect gas form a series of rectangular hyperbolas.

The work done by a gas during isothermal expansion or upon a gas during isothermal compression between any initial and final volumes, is found by integration of the area under the appropriate isothermal curve. The elemental work, dW = P dV.

Since $PV = P_1 V_1 = P_2 V_2 = MRT$, we have

$$W = \int_{V_1}^{V_2} P_1 V_1 \frac{dV}{V} = P_1 V_1 \log_e \frac{V_2}{V_1} = MRT \log_e \frac{P_1}{P_2}$$

Now, the internal energy change with change in volume of a perfect gas is zero at constant temperature; and since the heat added to any substance is the sum of the internal energy change and the external work done, we have for isothermal expansion of a perfect gas that the heat added to the gas is just equal to the work done by the gas. In other words, the efficiency of conversion of heat into mechanical work is 100 per cent. in isothermal expansion of a perfect gas. In order to secure isothermal compression of a perfect gas, it is necessary to remove from the gas an amount of heat exactly equivalent to the mechanical work done upon the gas.

Should it be necessary to calculate the work done by or upon a real gas during isothermal change with a greater degree of accuracy than afforded by the perfect gas relation, this may be accomplished by substituting for P in the expression for elemental work dW = P dV, its value in terms of V from one of the equations of state proposed for real gases, as given in a preceding article. This is rarely, if ever, necessary, however, because isothermal change is mainly assumed for ideal or theoretical conditions where the gas is considered to conform with the pressure-volume-temperature relation of a perfect gas. With a real gas, the heat added during isothermal expansion is not exactly equal to the work done because there is some change in interal energy with change in volume at constant temperature.

During isothermal expansion there is an increase in entropy of the gas and during isothermal compression there is a decrease in entropy equal to the heat added or abstracted divided by the constant absolute temperature of the gas. For the perfect gas relation, the heat added or abstracted is

$$\Delta Q = A\,W = MART \log_e \frac{P_1}{P_2}$$

The corresponding entropy change is

$$\Delta S = \frac{\Delta Q}{T} = MAR \log_e \frac{P_1}{P_2}$$

For one mole of gas, the entropy change is

$$\Delta S = 1.985 \log_e \frac{P_1}{P_2}$$

where the entropy change ΔS is in B.t.u. per lb.-mole per degree fahr. absolute or in cal. per gram-mole per degree cent. absolute, depending upon whether one lb.-mole or one gram-mole of gas is under consideration.

As an example of isothermal change, we will calculate the amount of heat that must be removed in the isothermal compression of 1000 cu. ft. of free air per minute from atmospheric pressure to 100 lb. per sq. in. gage in order to keep the temperature at 70 fahr. Substituting in

$$\Delta Q = AW = AP_1 V_1 \log_e \frac{P_2}{P_1}$$

we have $\Delta Q = \frac{144 \times 14.7 \times 1000}{778.6} \log_e \frac{100 + 14.7}{14.7}$

$$= 5711 \text{ B.t.u. per min.}$$

The entropy of the 1000 cu. ft. of free air is decreased by 5711 / (459.6 + 70) = 10.8 B.t.u. per deg. fahr. absolute.

Adiabatic Change

Adiabatic change, that is, expansion or compression of a gas without passage of any heat to or from the gas, is a phenomenon which is approached in many industrial processes. It is therefore customary to calculate the final conditions of a gas from the initial conditions under the assumption of adiabatic change and then apply corrections for deviations which have been found by experience to exist in similar situations. The theory of adiabatic change is thus of direct practical importance.

We will first discuss the case where the gas may be considered as having constant specific heats as well as conforming with the pressure-volume-temperature relation of a perfect gas. For the change in internal energy du in terms of k, we have

$$du = \frac{AR}{k - 1} dT.$$

Substituting in the general relation dq = du+Apdv and eliminating T by utilizing the pressure-volume-temperature relation of a perfect gas, pv = RT, we get

$$dq = \frac{AR}{k - 1} dT + Apdv$$

$$= \frac{AR}{k-1}\left(\frac{p}{R}dv + \frac{v}{R}dp\right) + A\,pdv$$

$$= \frac{Avdp + Akpdv}{k-1}$$

Since in an adiabatic change dq = O, we have

$$\frac{Avdp + Akpdv}{k-1} = O \text{ or } \frac{dp}{p} + k\frac{dv}{v} = O$$

whence $\log_e p + k \log_e v = \log_e C$ and $pv^k = C$

The temperature-pressure and temperature-volume relations for adiabatic change may be derived by combining $p_1 v_1^k = p_2 v_2^k$ with the relations $p_1 v_1 = RT_1$ and $p_2 v_2 = RT_2$, obtaining

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} = \left(\frac{v_2}{v_1}\right)^{k-1}$$

The expression for the work done during adiabatic expansion may be obtained by integrating the elemental work dW = PdV between the limits of V_1 and V_2 after substituting P from $PV^k = P_1 V_1^k = P_2 V_2^k$. There is thereby obtained

$$W = \frac{P_1 V_1 - P_2 V_2}{k-1}$$

which corresponds to the expression previously derived for the change in internal energy of a perfect gas. This is as it should be because in adiabatic expansion the work done is at the expense of the internal energy of the gas.

As an example of an adiabatic change, we will calculate the final temperature of air which is compressed from atmospheric pressure to 100 lb. per sq. in. gage, the initial temperature being 70 fahr. Instead of using the above temperature relation directly, we will first calculate the final volume for an initial volume of 1000 cu. ft. of free air. Substituting in $P_1 V_1^k = P_2 V_2^k$, we have

$$14.7 \times 1000^{1.4} = 114.7 \times V_2^{1.4}$$

or V_2 = 230.5 cu. ft. final volume

The final temperature may now be calculated by multiplying the initial temperature by the ratios of the initial and final pressures and volumes:

$$T_2 = (459.6+70) \times \frac{114.7}{14.7} \times \frac{230.5}{1000} = 952.6 \text{ fahr. abs.}$$

$t_2 = 952.6 - 459.6 = 493.0$ fahr. final temperature after compression.

For adiabatic change with large temperature variation, such as occurs in internal combustion engines, it is not sufficiently accurate to use the relations derived above for constant specific heats. It is in general permissible to use the pressure-volume-temperature relation of a perfect gas, pv = RT, but in the general energy equation, dq = du + A p dv, there must be substituted for du, the relation

$$du = (a_v + b\,T + e\,T^2 + \text{etc.})\,dT$$

because the specific heat at constant volume must be regarded as a function of the temperature T instead of being constant. Making this substitution and taking dq = O for an adiabatic change, we have

$$O = (a_v + b\,T + e\,T^2 + \text{etc.})\,dT + A\,p\,dv$$

Substituting the value of p from the perfect gas relation and dividing by T, we have

$$O = \left(\frac{a_v}{T} + b + e\,T + \text{etc.}\right) dT + AR\frac{dv}{v}$$

Integrating between limits, we obtain for the relation between temperature and volume,

$$a_v \log_e \frac{T_2}{T_1} + b\,(T_2 - T_1) + \frac{e}{2}(T_2^2 - T_1^2) + \text{etc.} =$$

$$AR \log_e \frac{v_1}{v_2}$$

To get the relation between temperature and pressure, we utilize the perfect gas relation

$$\frac{v_1}{v_2} = \frac{T_1}{T_2}\,\frac{p_2}{p_1} \text{ whence } \log_e \frac{v_1}{v_2} = \log_e \frac{T_1}{T_2} + \log_e \frac{p_2}{p_1}$$

$$\text{and } (a_v + AR) \log_e \frac{T_2}{T_1} + b\,(T_2 - T_1)$$

$$+ \frac{e}{2}(T_2^2 - T_1^2) + \text{etc.} = AR \log_e \frac{p_2}{p_1}$$

Since $a_p = a_v + AR$, we have

$$a_p \log_e \frac{T_2}{T_1} + b\,(T_2 - T_1) + \frac{e}{2}(T_2^2 - T_1^2) + \text{etc.} =$$

$$AR \log_e \frac{p_2}{p_1}$$

To obtain a more convenient form for calculating T_2 when T_1 is known, we can divide the last expression above by 2.3026 a_p, obtaining

$$\log T_2 = \log T_1 + \frac{AR}{a_p} \log \frac{p_2}{p_1}$$

$$+ \frac{b}{2.3026\,a_p} T_1 + \frac{e}{2 \times 2.3026\,a_p} T_1^2 + \text{etc.}$$

$$- \frac{b}{2.3026\,a_p} T_2 - \frac{e}{2 \times 2.3026\,a_p} T_2^2 - \text{etc.}$$

This equation is solved by assuming various values of T_2 until an identity is obtained.

Thus, let us calculate the temperature attained by adiabatic compression of air from 70 fahr. and atmospheric pressure to 100 lb. per sq. in. gage. For one mole of air, AR = 1.985, a_p = 6.93, b = O and e = 0.1200 / 10^6; also, T_1 = 70 + 459.6 = 529.6. By assuming various values of T_2, we finally find an identity with T_2 = 944.0 fahr. absolute, whence the temperature after adiabatic compression is 944.0 — 459.6 = 484.4 fahr. For constant specific heats, the final temperature was previously calculated to be 493.0 fahr.

Polytropic Change

Actual thermal changes during expansion or compression of real gases in industrial processes are in

general neither isothermal nor adiabatic but usually intermediate. Thus, in compressing air, the relation between the pressure and volume is neither

$$P_1 V_1 = P_2 V_2 \text{ nor } P_1 V_1^k = P_2 V_2^k$$

but may be approximately represented by

$$P_1 V_1^n = P_2 V_2^n$$

where the exponent n is taken as constant and generally has a value intermediate to 1 and k. Such expansion or compression has been called polytropic by Zeuner who derived it under the assumption that the quantity of heat supplied to or withdrawn from the gas is directly proportional to its change in temperature. The course of polytropic curves for air with $n = 1.1$, 1.2 and 1.3 are compared in Fig. 2 with the isothermal and adiabatic curves for $n = 1$ and 1.4 respectively. These curves become straight lines if plotted on logarithmic cross-section paper as in Fig. 3.

It may be shown that in a polytropic change, the work done by the gas during expansion or done upon the gas during compression is given by

$$W = \frac{P_2 V_2 - P_1 V_1}{n - 1}$$

The change in internal energy is given by

$$\triangle U = \frac{P_2 V_2 - P_1 V_1}{J(k - 1)}$$

We therefore obtain for the heat added or abstracted,

$$\triangle Q = \frac{P_2 V_2 - P_1 V_1}{J(n - 1)} + \frac{P_2 V_2 - P_1 V_1}{J(k - 1)}$$

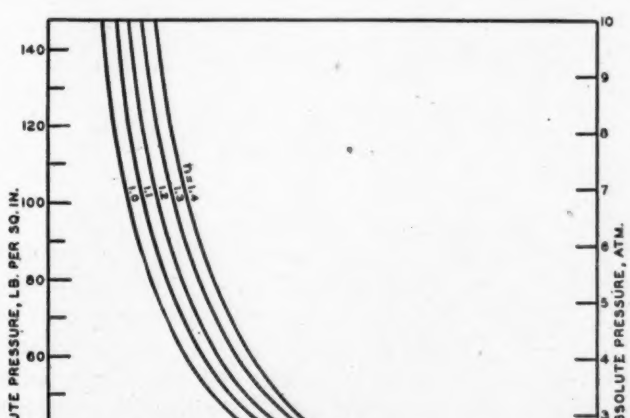


Fig. 2—Polytropic compression of air from atmospheric pressure

The change in temperature during polytropic expansion or compression is given by

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} = \left(\frac{V_2}{V_1}\right)^{n-1}$$

Let us calculate the amount of heat removed from 1000 cu. ft. of free air in compressing it from atmospheric pressure and 70 fahr. to 100 lb. gage with $n = 1.2$. Substituting in $P_1 V_1^n = P_2 V_2^n$, we have $14.7 \times 1000^{1.2} = 114.7 \times V_2^{1.2}$; whence $V_2 = 180.5$ cu. ft.

Substituting in $W = \frac{P_2 V_2 - P_1 V_1}{n - 1}$, we have

$$W = \frac{144}{1.2 - 1} (114.7 \times 180.5 - 14.7 \times 1000)$$

$= 4{,}322{,}400$ ft. lb. $= 555.2$ B.t.u. work done upon air.

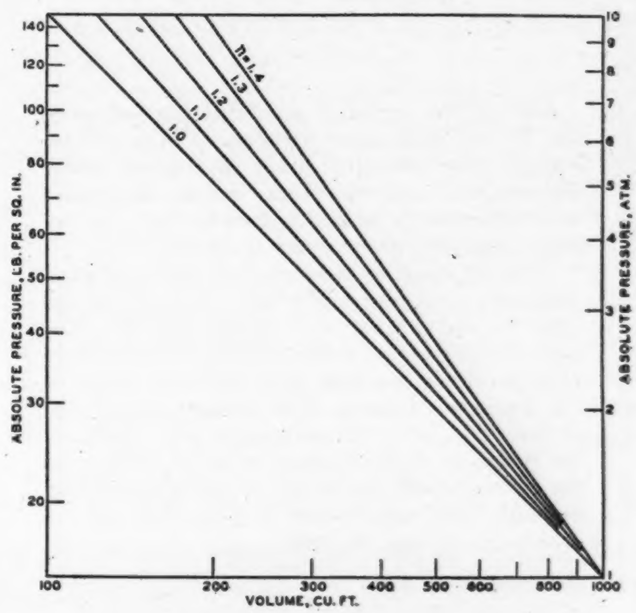


Fig. 3—Polytropic compression of air from atmospheric pressure (logarithmic co-ordinates).

Substituting in $\triangle U = \frac{P_2 V_2 - P_1 V_1}{J(k - 1)}$, we have

$$\triangle U = \frac{144}{778.6(1.4 - 1)} (114.7 \times 180.5 - 14.7 \times 1000)$$

$= 277.6$ B.t.u. increase in internal energy.

Whence, $555.2 - 277.6 = 277.6$ B.t.u. heat removed during compression.

References

The International Critical Tables, Partington and Shilling's book on The Specific Heats of Gases and Bulletin No. 139 of the University of Illinois Engineering Experiment Station by Goodenough and Felbeck entitled "An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous and Liquid Fuels" may be referred to for data on the thermal properties of other gases than those given in this article. Methods of utilizing such thermal data are discussed in the latter two references as well as in Goodenough's book on the Principles of Thermodynamics and in Zeuner's book, translated by Klein, on Technical Thermodynamics.

Cooperative Power Production in the Iron and Steel Industries of France

By

DAVID BROWNLIE

LONDON

During the period of rehabilitation following the World War, many mergers of men and resources were brought about in Europe under government control. Thus major economies were effected in many industries by the coordination of interests and facilities.

Two outstanding examples of the correlation of power resources under the direction of the state are the German developments in the Ruhr and the French program of cooperative power production in the iron and steel industries of the Lorraine district. The present article describes this latter development and indicates the progress that has been made in improving the economy and reliability of power generation through the coordination of the combined resources of a basic industry.

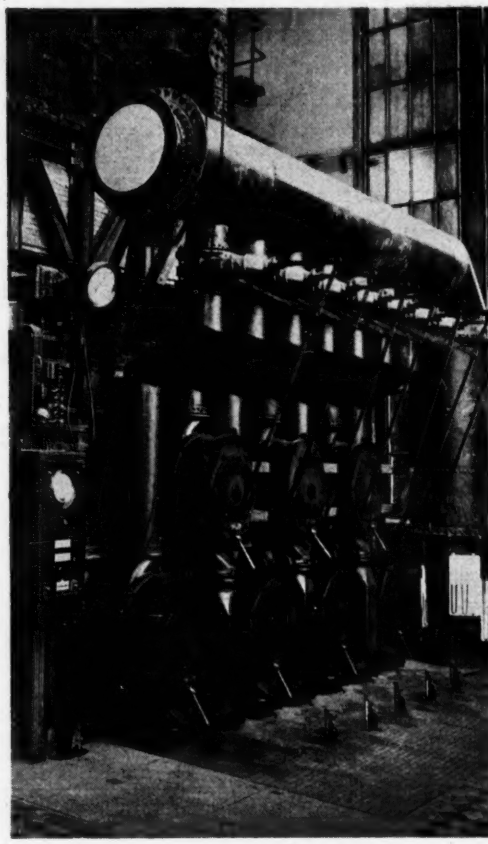


Fig. 1—Boiler fired by blast furnace gas and coke oven gas in plant of Société Electrique de la Sidérurgie, Lorraine.

SINCE the end of the World War in 1918, one of the most important developments in the field of combustion and fuel technology, is the grouping together of different mines, iron and steel works, and coke oven plants, for power production, with sale of surplus energy in the form of either electricity or gas. This efficient principle, of the greatest importance both from the industrial and the national point of view, has of course been facilitated by the strides made in overhead high-tension wiring, and also in the long-distance, high-pressure transmission of gas from coke ovens or other sources, as well as natural gas.

In Europe, the striking example of this development is the Ruhrgas A.G. and the construction, especially during the past few years, of hundreds of miles of mild steel pipe lines, which already supply a considerable proportion of the whole of Germany with coke oven gas as towns supply. The situation is peculiar in the sense that nearly all the coal required in Germany for the manufacture of metallurgical coke, about 40,000,000 tons per annum, is coked in the Ruhr. Consequently, the rich gas production in this area is about four times that of the whole of the German towns gas industry. Since about 1922, the coke oven plants of the Ruhr have been revolutionized by the wholesale scrapping of old types of ovens and the installation of new designs of high temperature ovens concentrated in a comparatively few very large plants.

The latest tendency, now being adopted more and more, is to fire the ovens with low-grade gas, such as blast furnace gas and producer gas made from coke breeze and small coal, so that the rich coke oven gas may be marketed. Practically all of the mines, iron and steel works, and coke oven works in the Ruhr are cooperating along these general lines with sale of the surplus coke oven gas to scores of towns, and extending service farther and farther into Central Germany.

It will be remembered that the vast high-pressure, long-distance pipe line systems for Natural Gas,

originated in the United States about 1891, although the first short pipe line with low-pressure dates back to 1872 in Pennsylvania.

France also has developed, to the fullest extent, the generation and distribution of electricity on cooperative lines, not only in connection with mines and the iron and steel industries, but also in water power. Hydro-electric plants and equipment to the extent of 2,500,000 hp. have been erected.

It is the purpose of this article to give a brief description of the highly interesting work that has been undertaken in the Lorraine district in France, especially since 1924, in connection with the co-operative generation and sale of superfluous electricity from the Lorraine steel works, thus utilizing to the fullest extent and on the most scientific lines, all the surplus blast furnace gas, coke oven gas, and small coke and coal for the generation of electricity, with the sale of the surplus, which at the present time nominally corresponds to about 100,000 kw.

There are in operation overhead transmission lines which inter-connect through various sub-stations ten different power stations at the various steel works, having a total output of about 125,000 kw. of which about 66 per cent is now developed by large blast furnace gas engines. At the same time, more and more modern boiler plants, fired with blast furnace gas and coke oven gas, are also being erected, while in some cases individual boilers are equipped for pulverized fuel firing, using direct-fired systems, to supplement the gas firing. By the end of 1930 installations of this kind totalling about 166,000 kw. will have been installed.

At present about 74,000 kw. of steam-driven plant is in operation in seven different works, and the whole system is controlled from a central station by a load despatcher, using a telephone.

I am enabled to give this description by the courtesy of A. Evain (Directeur of the Société Electrique de la Sidérurgie Lorraine, of Nancy), as well as that of L. A. E. Sekutowicz of Paris, and T. J. Gueritte (Secretary of the British Section of the Société des Ingenieurs Civils de France in London). Certain of the information given herewith is taken from the joint paper by L. A. E. Sekutowicz and A. Evain, entitled "Electrical Plant and Network of the Société Electrique de la Sidérurgie Lorraine," which was presented at a joint meeting of the Institution of Electrical Engineers and the British Section of the Ingenieurs Civils de France, in London. For the interesting photographs supplied especially for this description I have to thank M. Evain, who gave a paper "Etude Economique de la Production de l'Energie Electrique dans les Usines Metallurgiques" before the World Power Conference at Berlin in June, 1930.

Most of the remarkable developments in France

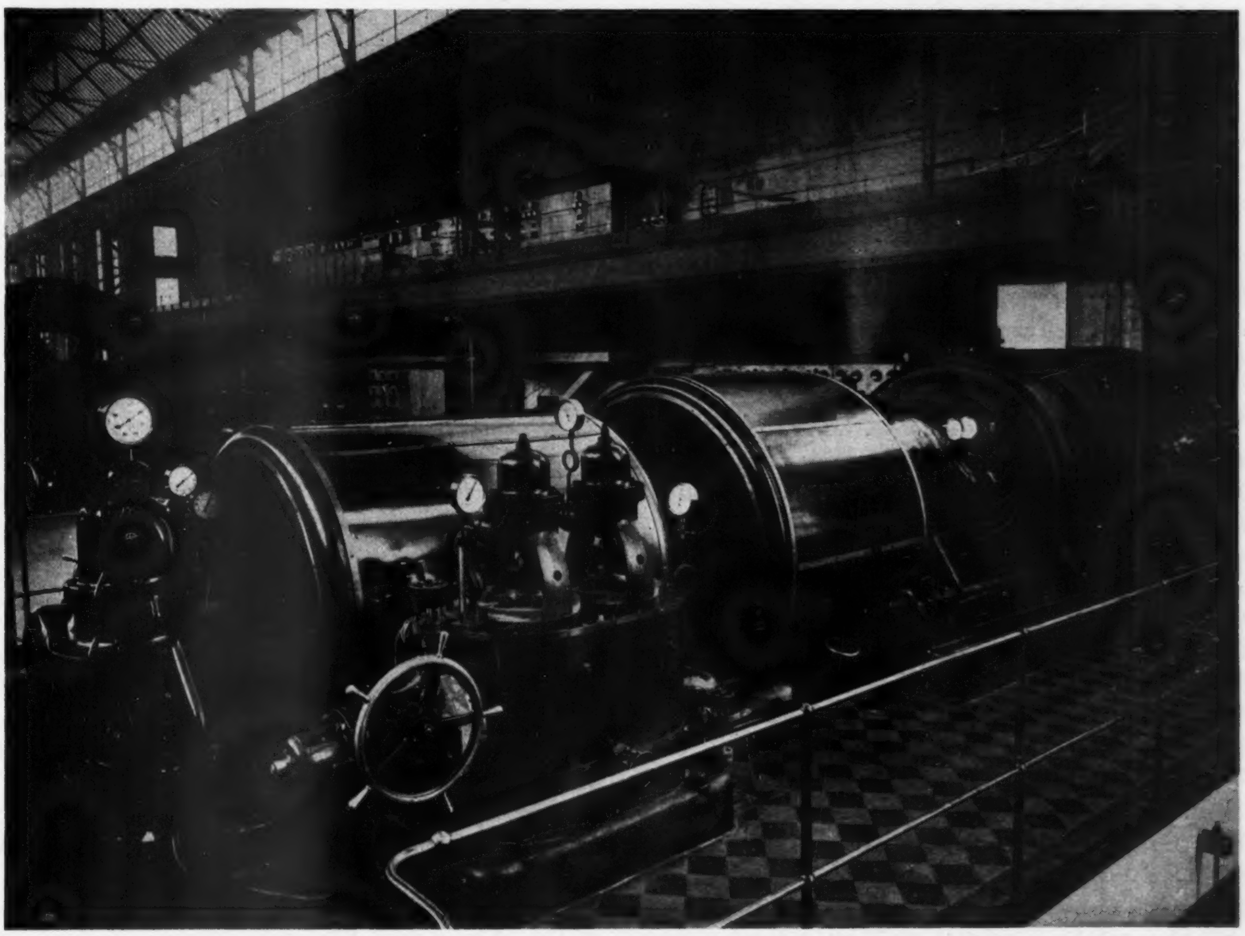


Fig. 2—Recently installed power station of Société Electrique de la Sidérurgie Lorraine, showing 15,000 kw. steam turbine in foreground and blast furnace gas engine in background.

in the field of fuel and power, including this particular work in Lorraine, were commenced after the War in 1918, and the Mining and Metallurgical Technical Commission of France devoted particular attention to the reconstruction of the mines and works devastated in the Lorraine area, particularly to developments in the overhead transmission of electricity. In this way, by means of ordinary coal-fired power stations, the iron and steel and other industries of Lorraine were enabled to come into operation again much more rapidly than would have been the case if they had had to depend upon themselves from the beginning.

Not including iron and steel works in Luxembourg and Belgium, the Société Electrique de la Sidérurgie Lorraine, which is a cooperative company directing the power development in Lorraine, now controls eight iron works with thirty blast furnaces manufacturing pig iron, and twelve complete steel works, including steel producing plants and rolling mills, having seventy blast furnaces, the total production averaging about 7,000,000 metric tons (2204 lb.) of cast iron per annum, and approximately 127,116,000 cu. ft. (3,600,000 cu. m.) of blast furnace gas per hour, of which about 50 per cent is used by the iron and steel industries direct and the remaining 50 per cent is available for the generation of electricity. Thus there is available for sale a supply which consists normally of 63,558,000 cu. ft. (1,800,000 cu. m.) of blast furnace gas per hour, and which is equivalent in energy to 240 tons of coal per hour with a heating value of 12,600 B.t.u. per lb. (7000 calories per kilo). Also this huge amount of blast furnace gas is equivalent to about 300,000 to 400,000 kw., depending upon whether it is used in blast furnace gas engines or for firing steam boilers and supplying turbines.

Up to the end of 1929, there had been completed the linking up of ten of the works, that is, sixty blast furnaces out of a total of one hundred, and in order to tackle this formidable problem the Société Electrique de la Sidérurgie Lorraine was founded in 1920 with a capital of 8,000,000 francs. There seems to have been some considerable delay in the matter in connection with the French Government, but the negotiations were all completed in 1924 and there was commenced the erection of an overhead transmission system of 65,000 volts, which links up with the French State Power Stations at Landres, as well as with various works in the French Ardennes and the Mines de la Houve.

As indicating the rapid growth of this scheme, the total power transmitted in 1923 was 7,931,074 kw., and in 1924, when the scheme began to come into full operation, the figure was 22,413,015 kw. This rose rapidly each year until in 1928 the output was 112,403,722 kw. and in 1929 had reached 130,000,000 kw. It is expected that by 1931 the figure of 200,000,000 kw. will be exceeded, and practically all of this vast amount of power is being produced by waste gas, chiefly blast furnace gas and coke oven gas, and supplemented—as already indicated—by pulverized fuel firing to take the peak loads.

Two highly complicated points are dealt with in the most lucid manner in the paper by Messrs. Sekutowicz and Evain, the first of which, however, relates to the different voltages in connection with the overhead transmission lines, and does not concern this article. The second, of direct interest to the combustion field, is the interesting point as to what proportions of the power generated by surplus blast furnace gas and coke oven gas should be utilized in large gas engines and in gas-fired steam boilers with turbines.

Essentially the principle adopted by the Société Electrique de la Sidérurgie Lorraine, following largely the experiences in other parts of the world, is to generate principally with large blast furnace gas engines, but at the same time to have a substantial portion of the energy generated by means of steam boiler and turbine plants, since in this way the fluctuations in the output of blast furnace gas are not so serious as would be the case with large gas engines only. Further, the great advance in pulverized fuel firing has tended to favor considerably the steam boiler and turbine, since it has been well demonstrated, for example, at the Fordson Plant of the Ford Motor Company, Detroit, that the whole of any available gas can be used to the fullest extent, irrespective of variations in the supply, with the balance made up by means of pulverized fuel firing, due to the extreme flexibility of this latter method. Also, important possibilities are presented for the use of waste heat boilers operated by the exhaust of the gas engines.

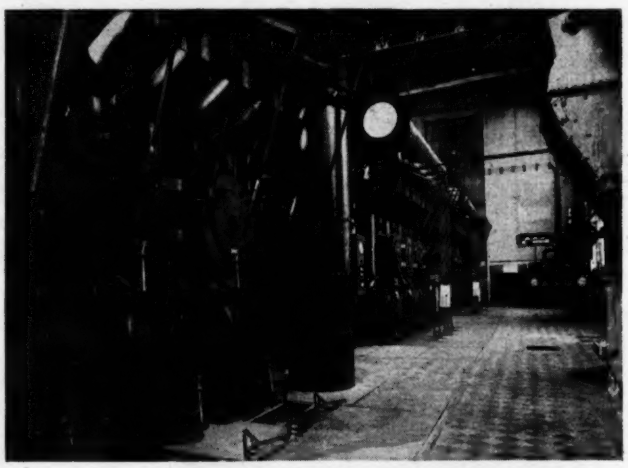


Fig. 3—Boiler room of Société Electrique de la Sidérurgie, Lorraine, showing three 498 lb. pressure boilers fired by blast furnace gas and coke oven gas, the boiler on the left also being equipped for burning pulverized fuel.

The exact condition in Lorraine at the end of 1929 was that 56 per cent of the total power was generated by means of gas engines, but the various additional steam plants now being equipped will still further reduce this figure to about 40 per cent

and the steam stations will then be generating the greater proportion of the power, about 60 per cent.

Many factors affect the relative economies of gas engines and steam plants in the converting of the available gas into power, but Messrs. Sekutowicz and Evain sum it up by stating that 3500 cu. ft.

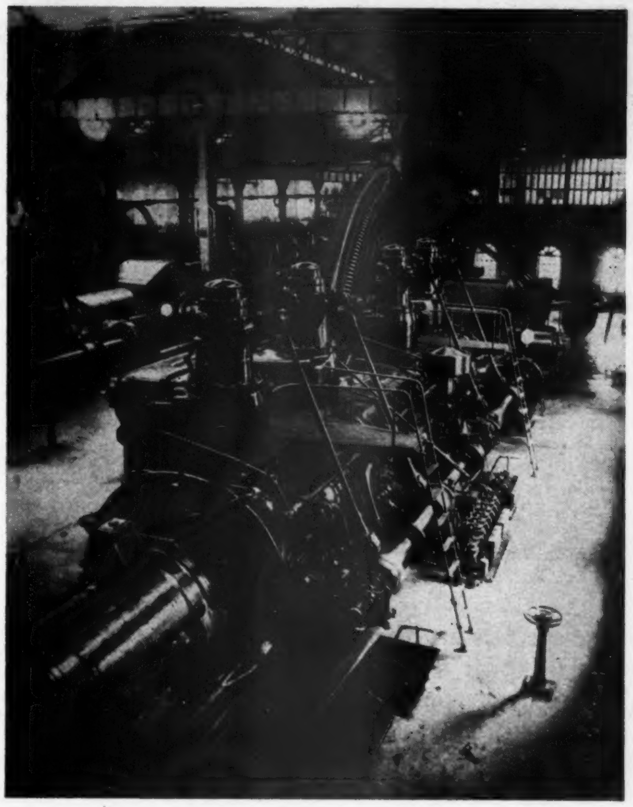


Fig. 4—3,000 kw. blast furnace gas engine in plant of Société Electrique de la Sidérurgie, Lorraine.

(100.0 cu. m.) of blast furnace gas will give 26 kw. with gas engines, and 21.7 kw. with steam boilers and turbines. That is, the gas engine shows an efficiency as regards actual power production of about 20 per cent more than that of the turbine plant. As against this, however, the gas engine is costly in manufacture and operation, and is not well adapted for dealing with the inevitable fluctuations in the supply of blast furnace gas. For this reason, the ideal arrangement, especially for such large schemes as that in Lorraine, is to combine blast furnace gas units and steam turbine units, so as to effect a compromise between the higher thermal efficiency of the gas engine and the superior flexibility of the steam plant, the exact proportions in each instance depending on the individual conditions.

A new factor has now come into the case, that is, the steam accumulator, which has the great advantage of storing up steam while acting as a reservoir or flywheel of energy, but the disadvantage is the very considerable capital cost, whether operated by the variable pressure or the constant pressure system.

A typical gas fired boiler installation, equipped for supplementary firing by pulverized coal, is shown in Fig. 1.

Fig. 2 is a view of the power station at the works of the Société des Aciepies de Micheville recently erected by the Société Electrique de la Sidérurgie Lorraine, showing at the back the blast furnace gas engine and in the front the 15,000 kw. steam turbine operated by three water-tube boilers, fired by blast furnace gas or coke oven gas as primary fuel. Fig. 3 is a view of the inside of the boiler house, showing the three boilers, each of 498 lb. per sq. in. pressure (35 kilos per sq. cm.). All three boilers are fired with blast furnace gas and coke oven gas, although the unit on the left is also equipped for pulverized fuel firing. The total output of the plant is 25 to 30 metric tons (2204 lb.) of steam per hour, the normal pressure at the turbine valve (Fig. 2) being 426 lb. per sq. in. (30 kilos per sq. cm.).

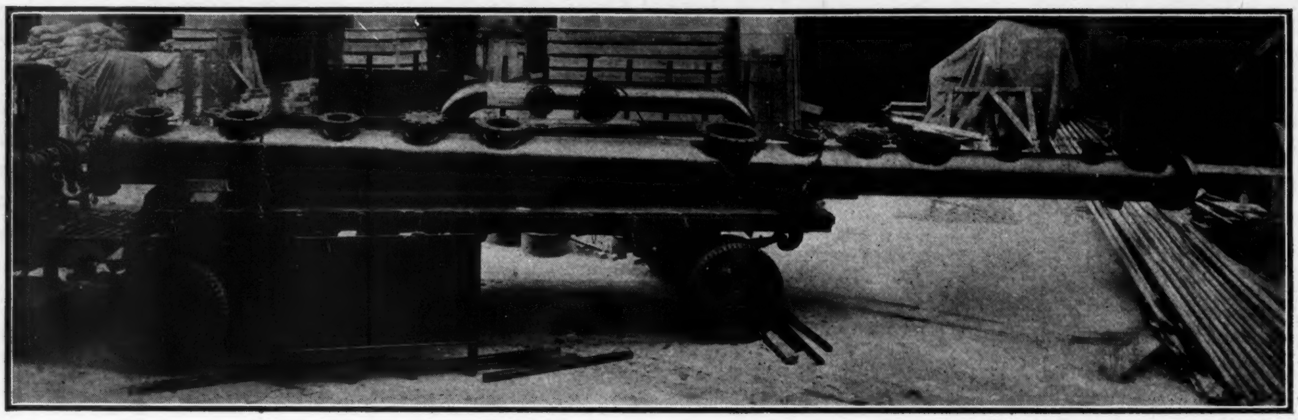
Two motor driven unit pulverizers supply the pulverized coal for supplementary firing. Each of these units has a capacity of from 2 to 5 tons of low-grade coal per hour. One of the large blast furnace gas engines of 3000 kw. capacity is shown in Fig. 4. By a slight adjustment, the burners used will take either blast furnace gas or coke oven gas equally well. The normal method of operation is to use all the blast furnace gas and then supplement this fuel, first with coke oven gas and then with pulverized coal.

The entire development is a fine example of the application of modern principles of power production in the iron and steel industries on co-operative lines. As the system is gradually being enlarged, it is necessary to take steps to distribute the surplus electricity over a wider and wider field, not only in Alsace-Lorraine but also in the Vosges area, while eventually the whole network is to be linked up through the French Ardennes with the Paris area and the North of France.

The Performance of Natural Draft Chimneys

(Continued from page 24)

perature have been decided upon and the draft-capacity and mechanical efficiency curves plotted, a natural draft chimney will operate along these curves in exactly the same manner that a centrifugal pump or a fan will operate along their curves as determined on test. Like a centrifugal pump or a fan, a natural draft chimney will do no more or less than that expected of it. Whether a chimney is used as the sole producer of draft or used in conjunction with a fan or blower, its performance will be the same. Finally, when a chimney is considered as performing the same identical service as a centrifugal pump or a fan, then an understanding of its performance will be more readily grasped and better control of this faithful draft producer may be had.



Oxwelded low pressure steam header, 39 ft. long and 20 in. diameter.

Oxy-Acetylene Welding for Power Plant Piping

By A. G. WIKOFF
Linde Air Products Company, New York

Oxy-acetylene welding is finding wide acceptance in the power plant field. Providing it is properly applied, there is much to be gained by its extended use. The author discusses its application in connection with power plant piping, describing the correct methods of making various welding operations and the procedure necessary to assure the best results.

ONE of the essential factors in the operation of the modern industrial or public utility power plant is an efficient piping system. Construction and designing engineers realize that not only must a piping system be economical and easy to install, but it must be of a type that will give the maximum efficiency and economy in operation. The demands of the operating engineer are brought to bear on the construction engineer. The present trend in power plant piping installations is turning more and more each day toward the use of oxy-acetylene welded construction because of its economy of installation and operation.

Construction Advantages

The advantages of oxy-acetylene welded construction for power piping are numerous. Considering them first from the point-of-view of the construction engineer, the economies of an oxwelded piping installation are apparent from the start, owing to the simplicity in the designing of systems for such construction. It is unnecessary to have special, odd-shaped fittings fabricated before the job is done as these specials may be made up right on the job from standard pipe by the use of oxy-acetylene cutting and welding apparatus. The flexibility of the oxy-acetylene process is another great advantage in the fabrication of power plant piping design. Changes in design during the actual fabrication will not tie up the construction work, as the erection engineer can have the pipe cut to fit with a cutting blowpipe and then welded in place so that the work can proceed without delay.

Another benefit to the construction engineer in the use of oxwelding is the reduction in weight of the transmission piping. The absence of heavy fittings or accessories necessary to other means of jointing makes for the easier handling of the whole job. The elimination of heavy fittings not only results in a reduction in material cost but also because of the decrease in weight lessens the difficulty of properly supporting the lines. By the elimination of these fittings, power plant piping may be placed where it would be impossible to make the installation were any other method used than oxy-acetylene welding. Welded piping may be easily installed in confined spaces, as in pipe shafts, in narrow tunnels or close to walls.

The fact that an oxwelded installation is the easiest to insulate is self-evident. The line presents a smooth uniform surface to which the insulation can be applied at minimum expense.

Operating Advantages

The advantages and economies of oxwelded construction in power plant piping are even more evident

to the operating engineer than to the construction engineer. The ideal system from the point of view of operation would be a continuous line without any joints. The nearest approach to this condition is the piping system with oxy-acetylene welded joints. There are no projections or uneven joints to interfere with efficient transmission. A joint is obtained, when properly made, fully as strong as the pipe itself and usually stronger. The strong and ductile joints produced by oxwelding are capable of withstanding the expansion and contraction stresses caused by the varying temperatures in the piping. The oxwelded joint is the most satisfactory joint for resisting corrosion, and will last as long as the pipe itself without the development of leaks. A leak-proof joint is one of the most desirable qualities of an oxwelded system, and the operating engineer need fear neither loss of power through the development of leaks nor any possibility of a shut-down for repairs with an oxwelded piping system.

The oxy-acetylene process is also the most effective method of making alterations on piping systems already in operation. These changes, moreover, can be made most readily on oxwelded lines. When it becomes necessary to place an additional branch connection in the line, the necessary outlet is cut according to templet with the cutting blowpipe and the new connection is welded in in a very short time so that the power shut-down is reduced to a minimum. The flexibility of the oxy-acetylene process is again an asset, when the question of additions to existing systems arises.

Procedure Control

The early hesitancy of power plant engineers to adopt oxwelding as a standard practice was in most cases based on a lack of confidence in the dependableness of the process. They would admit that oxwelding was a thoroughly sound and advantageous method when properly done but would always raise the question as to how they could be sure that the welds were all properly made. The establishment of procedure control welding during the past few years has applied the principles of production engineering to the welding process. Procedure control has afforded power plant engineers complete assurance of successful results with oxwelded construction and consequently has had a most important part in the development of welded steam piping. Uniform high-quality welds in all types of piping work are assured by oxwelding under strict procedure control methods.

The conditions which formerly caused varying results in welds have been studied, and a careful study of methods has revealed that proper control of the welding operation will guarantee uniformly satisfactory results.

Those who are responsible for power plant piping installations should be familiar with the principles of procedure control. The six requirements of procedure control have often been stated and they are again listed here as they form the important basis for any discussion of this subject:

A. *Check of the Welders*—to see that only competent operators are employed; further, that they prove their ability by successfully passing suitable qualification tests, preferably tests simulating the particular operation.

B. *Selection and Inspection of Material*—to make certain that only material suitable for welding is used and that proper welding rod is selected.

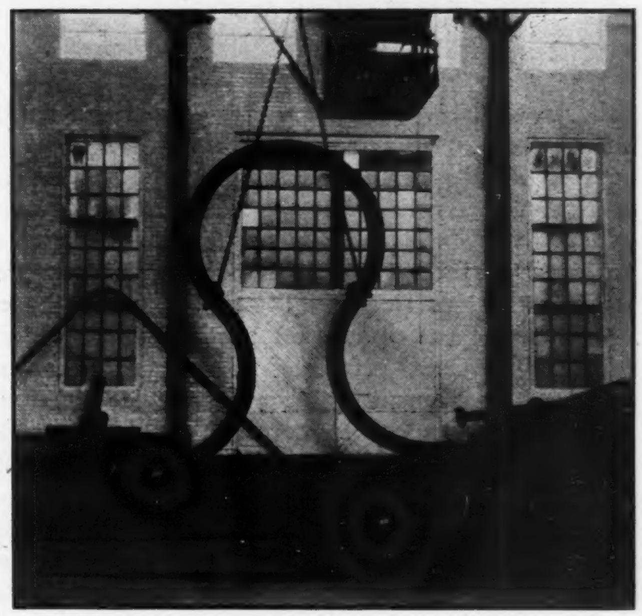
C. *Design and Layout of Welded Joints*—to insure that the joint is correctly designed for welding.

D. *Preparation for Welding*—to see that preparation and assembly are the best for welding.

E. *Organization and Welding Technique*—to see that the welding is carried on in the most satisfactory and economical manner and that the welding itself is correctly done.

F. *Inspection and Test*—to check whether the above items of the welding procedure have been followed and to see that proper welding results.

The test of the completed work is an additional assurance that the oxwelded installation will give satisfactory service.

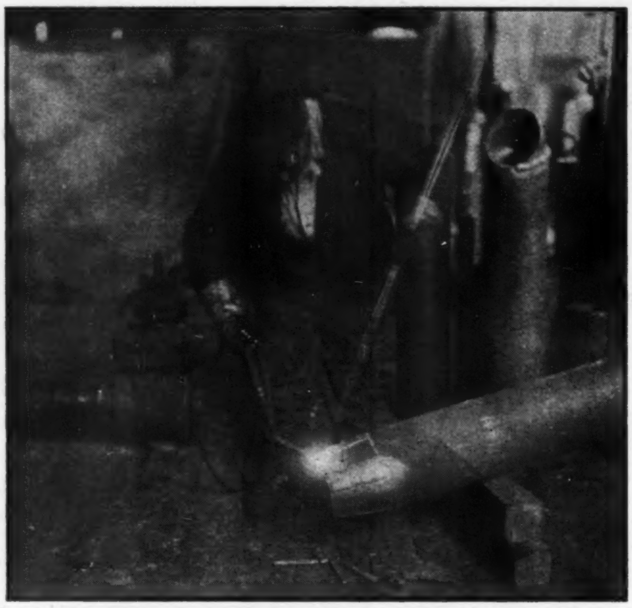


Oxwelded expansion bend in 12 in. steam pipe.

Procedure control methods have been adopted today for nearly all oxwelded high-pressure steam installations. With the above six factors as a guide, little difficulty is experienced by contractors in making oxwelded piping systems.

In order to enable welders to pass the qualification tests imposed by requirement A of procedure control, good training facilities have been established in many parts of the country. The increased use of oxwelding under procedure control has drawn attention to the importance of proper training for welders. The skill of a welder is no longer difficult

to determine, nor is it possible to make an error in determining a man's ability. Regular courses in welding are being given in trade and vocational schools everywhere. Some large public utility corporations have established schools within their own organizations to train men for pipe welding. Certain



Welding an orange peel reducer joint.

labor organizations provide a means of instruction in welding and encourage members of their trade to master the technique of this modern piping installation method. Many of the leading engineering schools have welding courses for the purpose of training engineers in the principles of oxwelded construction, particularly the management phases.

In most of the courses for training pipe welders, after the general principles of the oxy-acetylene process have been mastered, every effort is made to give the men a specialized training in piping work. In this way, competent welders are trained to do piping work and become experts in that line.

The check of a welder's ability under requirement A of procedure control does not depend upon mere statements of previous experience. It involves the welding of test specimens, which are the most accurate basis for determining the man's skill. In addition to the qualification test, periodic tests of the welder's work are made from time to time during employment. Completed test pieces are either sent to an approved laboratory for determination of tensile strength or are tested in a portable tensile machine on the job, and the results must equal or must exceed the minimum tensile strength specified in the procedure control. Ductility tests as well as tensile tests, are usually required. Various methods of making the tests are employed, depending upon the type of work to be done.

Under requirement B of procedure control comes the selection of the material and its rigid inspection. This involves the choice of pipe which is most suitable for welding. Just as important as the quality of the base metal is the use of the highest quality welding rod, which will insure a weld of maximum strength, which should be stronger than the pipe itself. In the selection of the welding rod, such qualities are considered as the ductility of the weld metal, ease of manipulation of the welding puddle, freedom from a tendency to burn, and the speed with which it may be deposited during the oxwelding operation.

Requirement C of procedure control calls for the correct design and layout for the use of oxwelded pipe joints. In the field of piping, comprehensive standards for oxwelded design have been developed by welding engineers. These standard designs were established to meet this procedure control requirement.

Under D, preparation and assembly, arrangements are provided to facilitate the use of oxwelding. In the case of large diameter heavy piping, the use of the proper clamps to align the piping and other jigs, which will save the welder's time by keeping the pipe in the proper position for welding, fall under this classification.

The supervision of the welders while the actual welding is being done comes under requirement E of procedure control. In the welding of steam piping it is very important that proper supervision be maintained in order to make certain that each welder constantly follows the correct practice for doing the work to which he is assigned.

6 in. and 8 in. risers from 8 in. main, 300 lb. steam line. Note the use of short radius bends.

The inspection and test provided for in requirement F furnishes the power plant engineer with an added assurance that the procedure outlined for the welding has been correctly followed and is successful. In the case of high-pressure steam piping, the procedure

control specification may require the oxwelded piping to be subjected to a hydrostatic pressure test of 1½ times the working steam pressure. While under pressure the welded joints are hammered with a 3 lb. hammer. This test provides absolute proof that the installation has been made according to specifications.

Line Welds

The single vee type butt weld is the type of line weld recommended for steam piping. The single vee butt weld when made under procedure control will consistently show a strength greater than the pipe itself and is the most economical type of weld to make. When properly made, there will be no protrusions on the inside wall of the pipe as this is easily controlled by any experienced operator. The ends of the pipe are beveled to 45 deg. with a shoulder, approximately 1/16 in. at the bottom of the vee. This allows complete penetration without any tendency toward the formation of protrusions on the inside wall.

Fittings and Specials

The fabrication of headers is one of the items of power plant piping in which the use of oxy-acetylene welding and cutting has been found to be especially advantageous. Large vacuum headers are fabricated entirely by the oxy-acetylene process and are installed as part of the exhaust steam system. High pressure headers are often made up as specials by pipe specialty companies and are regularly fabricated by means of oxwelding. Care must be taken in laying out pipe specials of this sort that the parts line up properly. The laying out of correct templets for the joints is of great importance in the fabrication of headers. Reducers are frequently necessary. These are fabricated by cutting vee-shaped pieces from the end of a piece of pipe, using suitable templets, so that the sections remaining will form the desired reducer after they have been swaged into contact and then welded. Oxwelded reducers are used in line construction as well as on headers. In much the same manner the end of a header or of a line may be closed by means of an oxwelded bull plug. The orange peel type of plug is fabricated in much the same manner as the reducer except that the sections, when swaged down and welded, close up the pipe entirely. In power plant pipe installations, it is necessary to use flanges on headers and on the line next to valves. In high-pressure installations, where flanges are necessary, a forged steel flange with a long neck designed for oxy-acetylene welding is recommended. These flanges have regular 45 deg. bevel for oxwelding and are fabricated in all standard sizes so that they may be welded to any standard size pipe. The necks of these forged steel flanges are of sufficient length to permit welding without any warping of the flange from the welding heat.

For bends in steam piping, specially fabricated short radius bends are recommended. These have come into wide use within the last few years and may be seen in many of the recent large steam piping installations. These bends consist of seamless steel tubing hot-formed by machine a special mandrel to give curves having a radius of only 1½ times the diameter of the pipe. These short radius bends are designed specially for oxy-acetylene welding and are furnished with beveled ends. The usual bend is a 90 deg. elbow, but this may be altered to make a bend of any size by cutting off enough of the turn to make the required elbow. Where the cut is made, the end will be a perfect circle and can be joined with one weld to the straight line.

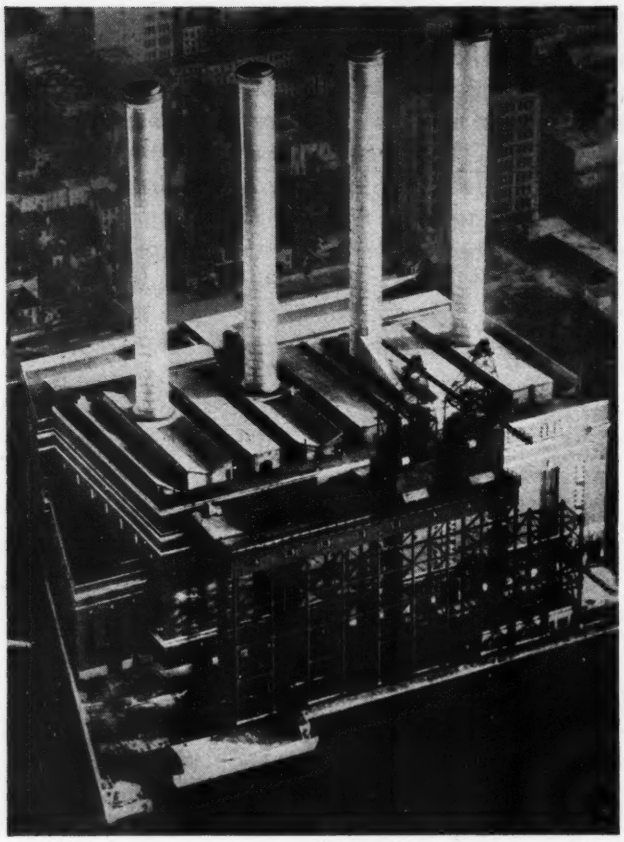
Drip pocket in steam line, fabricated entirely by oxwelding.

Branches from the main line are readily made by oxy-acetylene cutting and welding. The most common form of branch in a steam line is the tee, at right angles to the main pipe. Both the main and the end of the branch piece of pipe are cut with the blowpipe according to a carefully made templet and the edges are properly beveled subsequently. The branch is tack-welded in place and then welded. A branch at any angle to the main line may be installed by the oxy-acetylene process without difficulty, provided the templets for the cutting are carefully drawn before the actual work is started. Branches at almost any angle are a common sight in oxwelded piping systems.

Many other pieces of steam piping fittings are fabricated by oxy-acetylene welding. Such items as steam receivers and drip pockets are most easily fabricated by this method.

The constantly growing popularity of the oxwelded joint for power plant piping is based on economies extending from the inception of the system throughout its useful life. The outstanding feature of oxwelded construction is the permanency and tightness of the joints which insures maximum operating efficiency and practically eliminates maintenance.

Brooklyn Edison Places Large Boiler Contract



Hudson Avenue Station

DURING the past month, Matthew S. Sloan, President of the Brooklyn Edison Company and associated companies, announced that a contract for eight new boiler units, amounting to over $2,000,000, had been awarded to Combustion Engineering Corporation, New York. This is said to be the largest single contract for boiler equipment ever placed in this country. A sectional elevation of two of the new units is shown on Page 45.

The boilers are to be fired by multiple retort underfeed stokers, the contract for which, amounting to approximately $500,000, has been awarded to the American Engineering Company, Philadelphia.

In addition to the boilers, the contract awarded to Combustion includes complete water-cooled furnaces, boiler settings and steel work, and economizers.

The boilers are of a special bent-tube design, containing 24,540 sq. ft. of heating surface each and equipped with Elesco superheaters. Each boiler unit is guaranteed for a maximum evaporation of 530,000 lb. of steam per hr. with a maximum moisture content of 1 per cent. The design pressure is 500 lb. per sq. in. with an operating pressure of 440 lb. at the superheater outlet. The total steam temperature is 750 deg. fahr.

The two upper drums and the lower drum of each boiler are 54 in. dia. and 2½ in. shell thickness. There is also a dry drum of 48 in. dia. and 2 in. shell thickness. The distance between the upper and lower drum centers is 29 ft. The boiler tubes are to be arranged in three banks, the first bank being 39 tubes wide and 6 tubes deep. These tubes will be spaced on 7½ in. centers to permit the installation of a 114 element superheater. The second tube bank will be 48 tubes wide and 8 tubes deep on 6 in. centers, and the third bank 48 tubes wide and 7 deep, also on 6 in. centers.

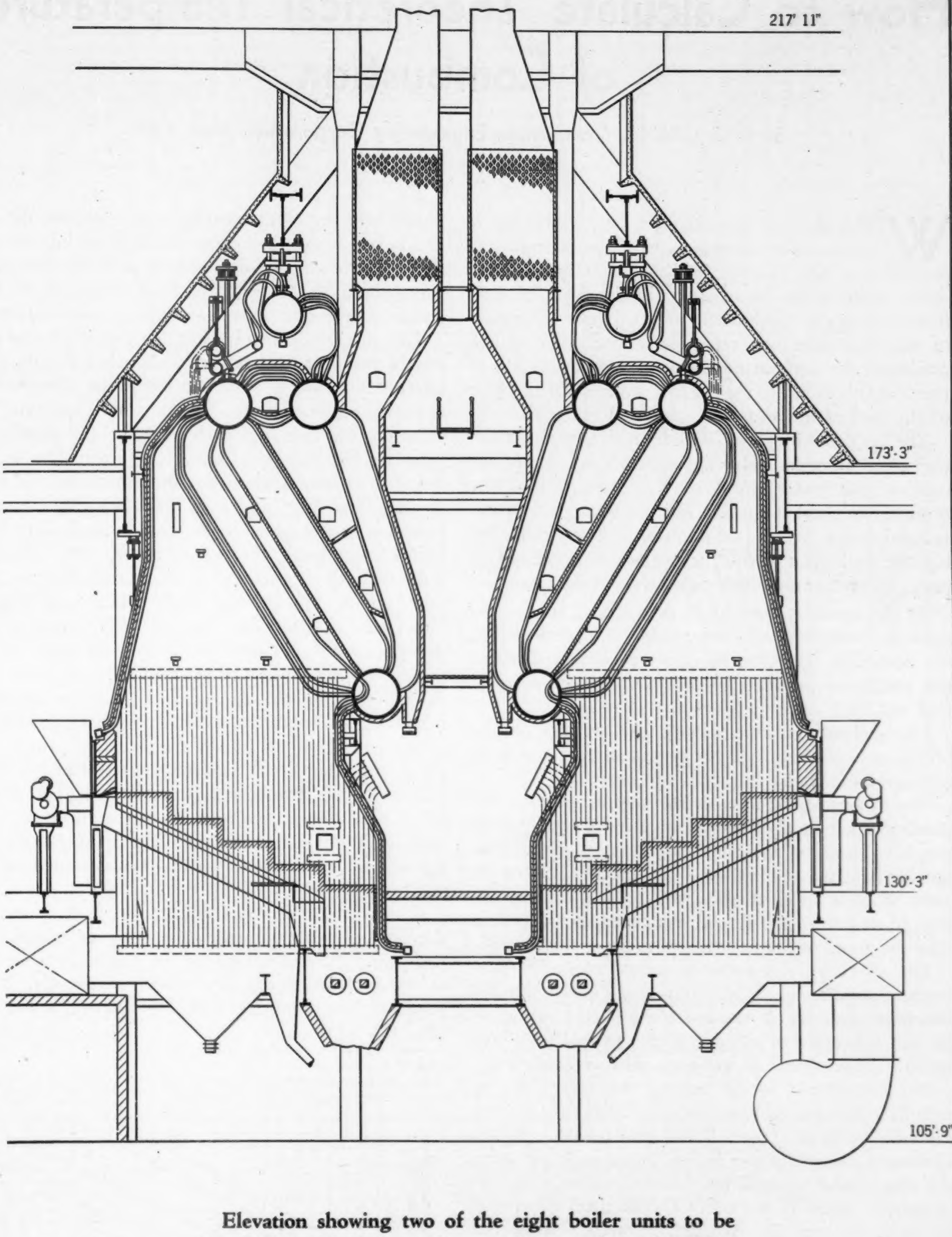
These boilers will serve two 160,000 kw. turbo-generators, contract for which was recently placed with the General Electric Company. These will be the largest single-shaft turbo-generators ever built.

The tubes comprising the front and rear walls and the two side walls are of the finned type. There will be 48 tubes in the rear wall, 49 in the front wall and approximately 51 tubes in each side wall. Risers from the side wall screens are entirely within the furnace leading from the upper side screen headers to the upper drums. This arrangement affords protection for the boiler walls above the furnace proper, provides additional heating surface and reduces insulation costs. The side wall tubes are covered with iron protecting blocks, cast onto the tubes, for a short distance up from the tuyere line of the stokers. The rear wall tubes also are covered by protecting blocks for a distance of about 6 ft. above the clinker grinders. The total heating surface of the furnace wall tubes is 3750 sq. ft., and the furnace volume of each unit is 15,500 cu. ft.

The stokers will be single units approximately 26 ft. 6 in. wide, 27 ft. deep, and will be the longest stokers ever built. Each stoker will be capable of burning approximately 56,000 lb. of coal per hr. at maximum rating. The control of the fuel feeding mechanism will be by electric motors, and the coal pusher mechanism will be adjustable over the entire width and length of each stoker. The clinker grinder rolls will be 36 in. in diameter.

The economizers are of the fin tube, return bend type and are designed for 650 lb. pressure. Each economizer will be 29 tubes wide and 32 tubes high, each tube having an effective length of 21 ft. The heating surface per economizer is 22,400 sq. ft.

A notable feature of this installation is that the general design provides a maximum of heating surface and furnace volume within the building space available.



Elevation showing two of the eight boiler units to be installed in the Hudson Avenue Station of the Brooklyn Edison Company. All eight units are of identical design throughout. Four are scheduled to be in operation by August 1931 and the remaining four by March 1932.

How to Calculate Theoretical Temperatures of Combustion

By B. J. CROSS, Combustion Engineering Corporation, New York

WHEN fuel is burned, the heat liberated is contained in the products of combustion. A part of this heat is contained as latent heat in the water vapor from the moisture of the fuel and that formed from the combustion of hydrogen. This part of the heat does not raise the temperature of the products. In computing flame temperatures, it is therefore necessary to use the low or net calorific value of the fuel rather than the gross calorific value.

The net calorific value of a fuel is determined from the gross calorific value by subtracting the latent heat of the water vapor that is contained in the products of combustion. The net value, therefore, is equal to the gross value minus $9\,H_2 \times 1050$, H_2 being the hydrogen in the fuel as burned expressed in parts by weight or per cent divided by 100.

As the net heat content so determined all goes to raise the temperature of the products of combustion, the resultant temperature may be computed when the weight of products of combustion per pound of fuel and their specific heats are known.

The method of determining the weight of the products of combustion per pound of fuel has been explained in Article No. 7 of this series.

The mean specific heat of the furnace gases is dependent upon their composition and the temperature range. The constituent that chiefly affects the specific heat is water vapor. The specific heat of all gases increases with temperature. The mean specific heat of gases of combustion, computed above 60 deg. fahr., is given in the curve, Fig. 1, for various fuels.

The chart on the opposite page gives the theoretical temperature of the products of combustion for different amounts of excess air, the latter being expressed in weight of gas per pound of fuel fired. An initial temperature of 60 deg. fahr. is assumed.

As an example in the use of this chart, let us assume a Pittsburgh bituminous coal having a gross calorific value of 13500 B.t.u. per pound. If the hydrogen content of the coal as burned is 5 per cent, the available heat will be $13500 - 1050\,(9 \times .05) = 13030$. With 13 per cent CO_2 the total gases will be 15.25 lb. per lb. of coal. Starting with the calorific value 13030 on the left hand vertical scale, trace horizontally to the line for 15.25 lb. of gas. Then trace vertically to the specific heat which may be taken from the chart, Fig. 1, at about the expected temperature — .270. From this last intersection, trace horizontally to the right hand vertical scale to final gas temperature — 3150 deg. fahr. By again referring to the specific heat chart we see that at 3150 deg. fahr., the specific heat should be .275 instead of .270. We may then repeat the procedure, using the more nearly correct value for specific heat and find 3100 deg. fahr. as the final temperature.

This final temperature is the theoretical value that would result if no heat were absorbed during combustion. Actually some heat will be absorbed by exposed boiler surfaces and a small amount lost through the furnace walls so that the final temperature will be lower than the theoretical value.

Let us assume further that the ash of the coal fuses at 2400 deg. fahr. and it is necessary that the temperature of the gases be reduced to 2200 deg. fahr. before they enter the boiler tubes. We therefore trace from the right hand vertical scale at 2200 deg. fahr. to the specific heat .267 taken from Fig. 1. From this intersection, trace to the pounds of gas per pound of coal and thence horizontally to the available heat content of the coal. To obtain a final gas temperature of 2200 deg. fahr., the available heat per pound of fuel must be reduced from 13030 to 8930, or we must absorb in the furnace 4100 B.t.u. per pound of coal fired. If the coal burning rate is 10,000 lb. per hr., the heat to be absorbed in the furnace will be 41 million B.t.u. per hour. If the exposed boiler surface is not sufficient to absorb this amount of heat, then additional water cooled surface must be added to make up the deficiency.

It may be seen from an inspection of the chart that the final gas temperature may also be reduced by increasing the amount of gas per pound of coal, that is, by increasing the amount of excess air. This, however, is a more expensive method as it lowers the efficiency of the unit.

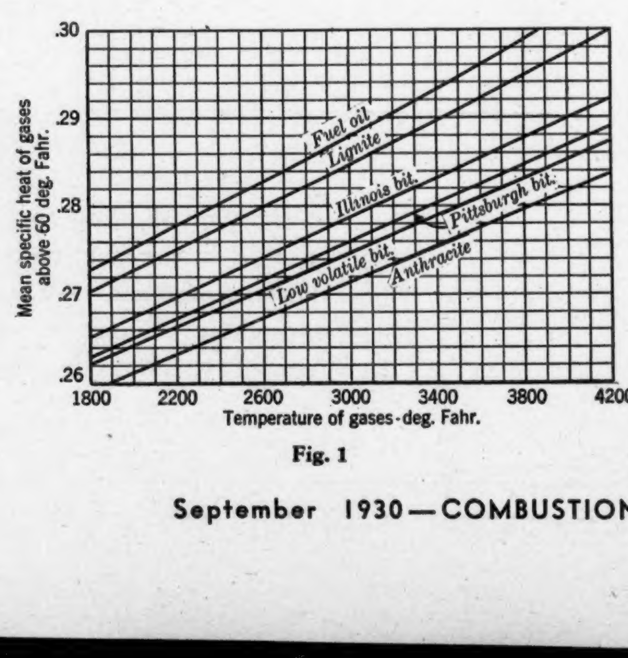


Fig. 1

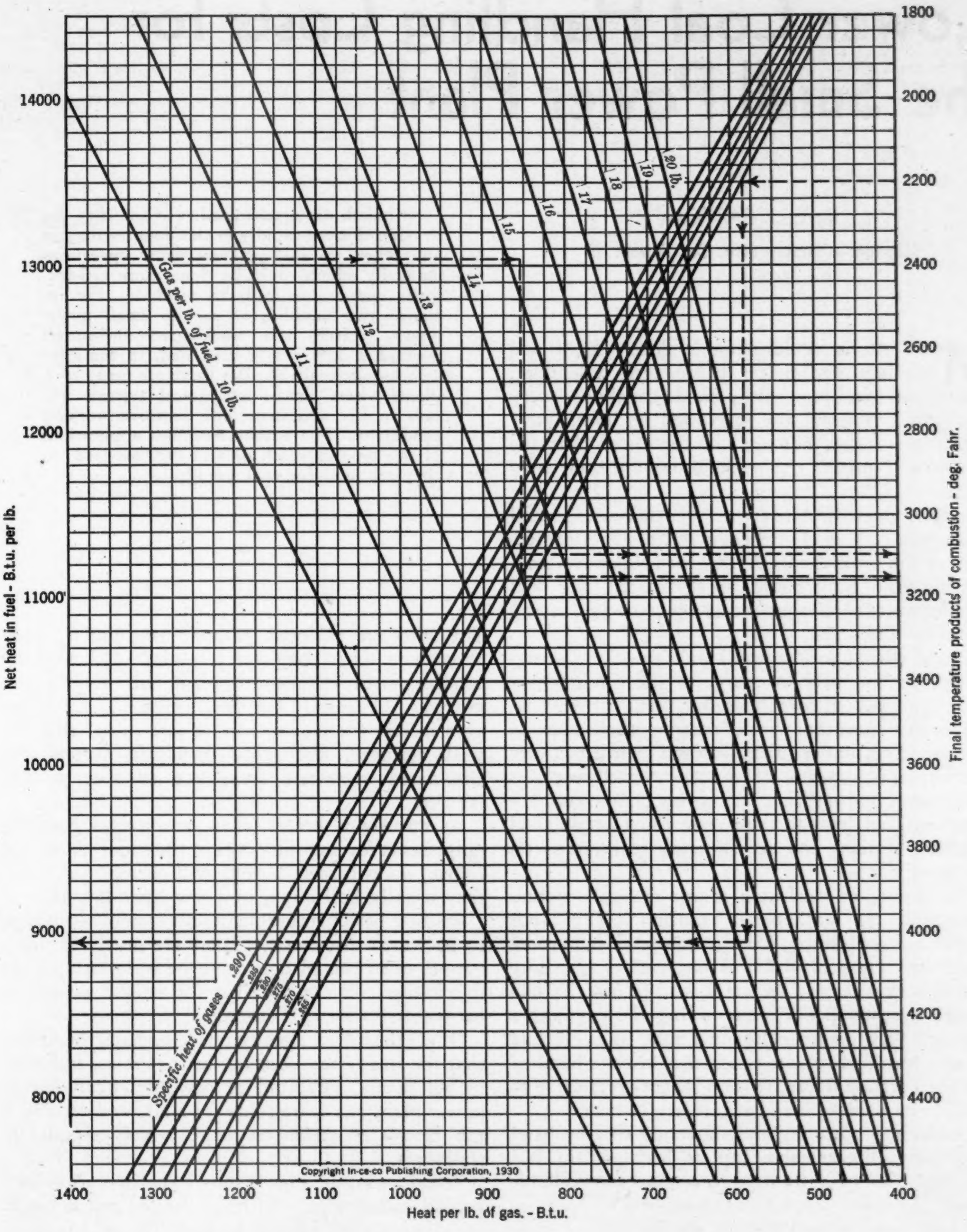


CHART FOR CALCULATING THEORETICAL TEMPERATURES OF COMBUSTION

No. 14 of a series of charts for the graphical solution of steam plant problems

Lower Coal Handling Costs for the Small Power Plant

By E. J. PATTON
Stephens-Adamson Mfg. Co., Aurora, Ill.

Although the mechanization of industry has developed more rapidly in this country than in any other, we still perform many operations manually which could be done more satisfactorily and economically by machine. For instance, the handling of coal in small power plants. Here, the wheelbarrow-and-shovel method is still widely used and, in some instances, advisedly so. But there are many such plants where simple mechanical systems will save money and materially improve working conditions. The author describes applications of this type in which silos of the barnyard variety are used for coal storage.

MOST of us could name a dozen small and medium size power plants still unloading, storing and handling coal by hand. In almost every case the operators are interested in mechanical handling systems, but are not convinced that the saving will justify the expenditure for their own plants.

In some cases they are right. In other cases careful study would show that there is at least one arrangement of conveyors by means of which the shovel-and-wheelbarrow gang could be eliminated, the coal handled quicker, cleaner and at a worthwhile saving in dollars and cents.

Often, when a mechanical handling system is suggested, the engineer in charge of the small power plant pictures a complicated system of feeders, conveyors, elevators, etc., costing thousands of dollars. As a matter of fact, several simple but very effective arrangements have been worked out and many small and medium size power plants formerly handling coal by hand are now using mechanical handling systems and are showing real profits on the investments.

The saving in hard cash is not the only recommendation for mechanical handling. The labor problem is simplified. Coal is unloaded and stored faster and cleaner, and the mechanical handling system makes it easy to weigh and record the fuel burned by each boiler, thus giving the engineer a definite check on his equipment that he never had before. And, of course, when the tonnage handled reaches a certain figure, it is no longer practicable to handle the coal by hand regardless of cost.

Unloading, wheeling and shoveling coal by hand is a dirty, disagreeable job at best and in bad weather it is especially hard to get and keep reliable labor. Almost every day coal must be wheeled from the yard into the boiler room, the doors have to be kept open for several hours at a time and in the cold winter months wind and dust mix, making the boiler room mighty unpleasant.

On the other hand, the mechanical handling system receives coal from the railroad car or motor truck, conveys and elevates to a storage bunker and then delivers to the stokers as the fuel is needed. The units are enclosed, manual handling is eliminated, the air is comparatively clean and working conditions in the power plant are greatly improved. Ordinarily but one operator is necessary, even in fair size plants, so it is safe to say that the well designed mechanical handling system has eliminated anywhere from one to four or five laborers.

As mentioned before, there are several good arrangements for handling coal in the small plant. Of course, no two installations will ever be alike, but they usually boil down to a few basic layouts.

Of these basic arrangements, one is unusual enough to be worth discussion. There are nearly as many variations of this plan as there are installations, but the feature common to all is the unique use of the barnyard silo for coal storage.

Of all the advantages claimed, perhaps the first is that the silo system seems to be the easiest to install in old plants. But old or new, practically every power plant has spare room for a silo and usually it can be placed so as to make a very simple installation.

The drawing in Fig. 2 shows the operation of a typical installation. Railroad cars or motor trucks are dumped over a track hopper. The coal is automatically fed to a bucket elevator which discharges into the top of a silo that has been doctored for the purpose.

Inside the top of the silo is a steel bin of about 25 tons capacity and small enough to allow a space between it and the inside of the silo. As coal is elevated to the silo, it fills the steel bin and finally

FIG. 1
Arrangement of various units for silo system. This installation requires but one 5 hp. motor.

FIG. 2
Typical arrangement of silo system for handling and storing coal in small power plants.

FIG. 3
Arrangement of double silo coal handling system with elevator installed between the silos.

FIG. 4
In this installation a belt conveyor carries the coal to the elevator and a screw conveyor delivers it to the bin.

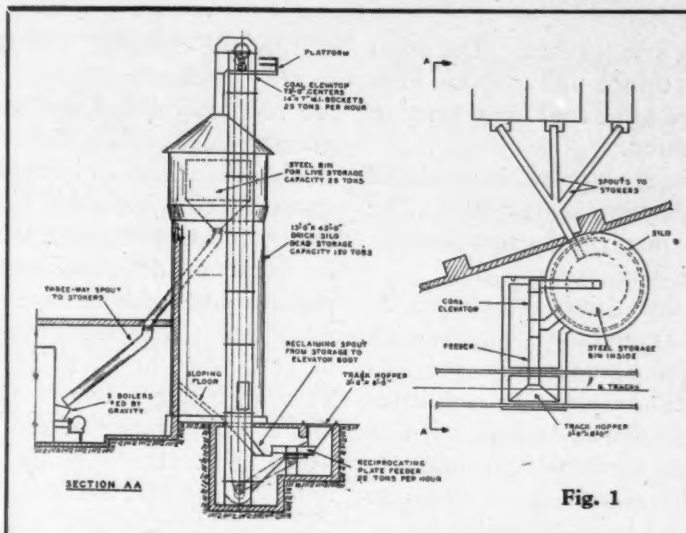


Fig. 1

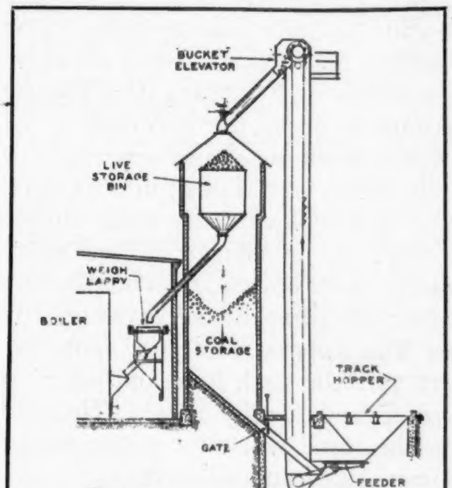


Fig. 2

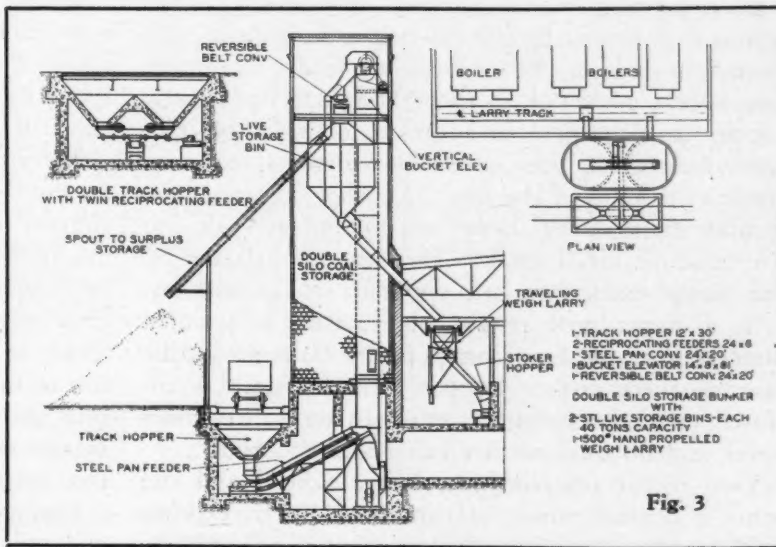


Fig. 3

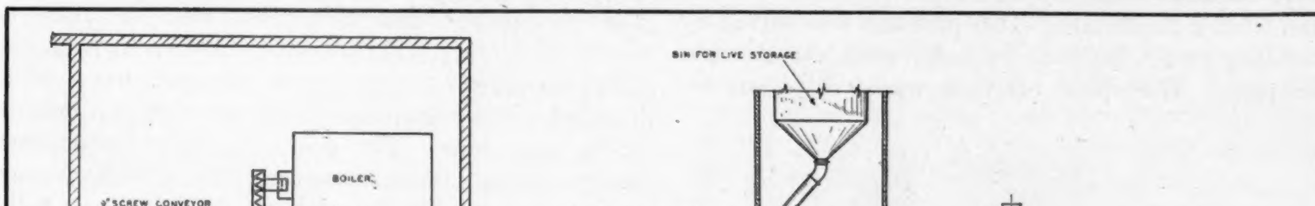


Fig. 4

overflows to the body of the silo beneath. The small steel bin above is for live storage and a spout from the bottom of the bin delivers the coal by gravity to the boiler room as it is wanted.

When the live storage bin is emptied, it is refilled by opening a gate in the bottom of the silo. The coal then flows by gravity into the bucket elevator and is re-elevated to the live storage bin.

In the smaller plants the coal can usually be fed by gravity to each stoker. Larger power plants with several boilers usually need some means for distributing the coal to different boilers. In such cases a traveling hopper or larry is generally used. If this is equipped with a scale the operator can not only distribute the fuel, but he can keep an accurate record of the amount burned by each boiler.

In equipping plants already in operation, the layout is governed by the existing arrangement. As shown in Fig. 4, the Milwaukee Chair Company uses a belt conveyor to carry the coal from track hopper to elevator. In addition, considerable elevator height has been saved by using a screw conveyor at the top of the silo. Another feature is the special distributing larry which operates on an overhead monorail track. This was necessitated by the curved track from silo spout to stoker hoppers.

In a paper mill power plant, coal is spouted directly from the live storage bin to boilers—in this case to three stoker hoppers. Every unit is enclosed, all transfer points between units are under cover and no dust or dirt can reach the plant.

Two recent installations, one a school and the other a central power station in a North Dakota city, required more storage capacity than could be had from a single silo. This problem was solved by building two silos, side by side, with the elevator between. The space between was walled in, enclosing the elevator, and the result was a well-protected, symmetrical unit. These two installations are both large, but the use of two silos permitted gravity feed to all boilers.

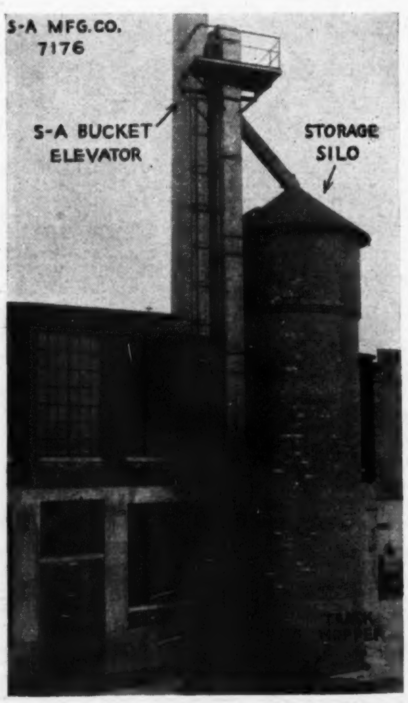


FIG. 5
Small silo installation providing 175 tons storage capacity.

The silo makes an ideal storage bunker, for a given ground area it has good capacity, it is reasonable in cost and the appearance is not bad. Single silos can be built in different diameters and heights, depending upon the storage wanted. In the ordinary small plant, operating two or three boilers, a silo of 13 ft. dia. by 35 to 45 ft. high is large enough. This will give a live or gravity storage of 25 tons and a dead storage of 100 to 150 tons. Larger silos up to 18 ft. dia. by 55 to 60 ft. high are practicable and will increase the live storage to 40 tons and the dead storage to 250 to 290 tons.

Where a larger storage capacity is needed, two or even more units can be built side by side. One elevator can be used to fill them all, some times by spout direct to each silo or by means of a shorter elevator feeding onto a distributing conveyor.

The silo system is suitable for nearly any small or medium size power plant. It is low in first cost, simple to operate and upkeep costs are almost negligible. The engineer in charge of one fairly large plant, operating three boilers, ran comparative tests before and after installing his silo system. By hand, it had cost him approximately 32 cents per ton to unload cars, pile the coal, load and wheel it into the boiler room and fire by hand. The silo system now does the same work for 3.8 cents per ton, including power, labor and all upkeep.

One power company operating electric lighting and power plants in Minnesota and North Dakota has now equipped four of its plants with silo systems for handling fuel. This company reports as follows: "Approximately 12,000 tons of coal have been handled since installation, the average per month being 1000 tons. The operating cost for current has been approximately one cent per ton with current at five cents per kw.-hr. We are receiving two inch lignite in hopper bottom gondolas and are handling it mechanically from cars to boilers. If handled by hand, the operation would be very costly."

There is no reason why either the original cost or the operation of such an installation should be prohibitive. Engineers experienced in designing coal handling installations can often suggest and lay out a mechanical handling system of this sort that will surprise the average power plant operator with its simplicity and possibility for savings.

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Cover Picture

The very unusual aerial photograph of New York, shown on this month's cover, was made by the Fairchild Aerial Survey, Inc., and is reproduced through the courtesy of Dwight P. Robinson & Company, Inc.

NEWS

Pertinent Items of Men and Affairs

Ceramic Engineer Joins Staff of Battelle Institute

Dr. Harold E. Simpson has been appointed a staff member of the Battelle Memorial Institute, Columbus, Ohio.

Dr. Simpson has had considerable experience in industrial work dealing with ceramics. He received both his undergraduate and graduate training at Ohio State University, where he specialized in ceramics. He comes to the Institute from Rutgers University, where he held the position of assistant professor of ceramics.

Dr. Simpson will work under the direction of Clyde E. Williams, assistant director, and R. A. Sherman, chief fuel engineer at the Institute, on a project dealing with the slagging action of coal ash on boiler furnace refractories. This investigation is to be made in cooperation with the American Society of Mechanical Engineers.

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The Foxboro Company, manufacturer of industrial instruments, Foxboro, Mass., announces several changes in its organization.

J. B. McMahon, former Tulsa branch manager, has been placed in charge of all field engineering service and direct sales and will make his headquarters at the home office at Foxboro.

G. B. Lane, former manager of the Detroit office, will succeed Mr. McMahon as branch manager at Tulsa.

A. W. Taber, former manager of the Atlanta office, has been transferred to Detroit and W. W. Barron has been appointed manager of the Atlanta office.

•

McClave-Brooks Company, manufacturer of fuel burning equipment, Scranton, Pa., announces the appointment of S. C. Orell as Chicago district manager. Mr. Orell was formerly district sales manager for the Murray Iron Works, Burlington, Iowa.

•

The Brown Instrument Company, Philadelphia, Pa., announces the retirement of George W. W. Cornman, Treasurer and Manager of the Service Department. Mr. Cornman has been in the employ of the Brown Company and associated interests for the past thirty-five years.

R. O. Kennan succeeds Mr. Cornman as Manager of the Service Department and E. T. Nahill has been made Field Supervisor.

Quigley Company Changes Name

The Quigley Furnace Specialties Company announces a change in the name of the company to Quigley Company, Inc., Originally this company dealt in furnace specialties only, but in recent years other products have been added, several of which are not related to furnace work.

Quigley products now include: Hytempite, high temperature cement; Q-Chrome, a neutral base refractory cement; Acid Proof Cement, for acid resisting masonry; Ganisand, granular refractory; Quigley Gun, for shooting refractory and concrete mixtures; Triple-A Protective Coatings for metal, concrete, stucco, wood, etc.; Q-Seal, a plastic, expansive, joint-sealing compound and Annite, an all purpose cleaning compound. The main office of Quigley Company is at 56 West 45th Street, New York City, with distributors in ninety cities throughout the United States and at thirty-five points in foreign countries.

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The American Refractories Institute will hold its fall meeting October 24, at the Traymore Hotel, Atlantic City, N. J.

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Babcock & Wilcox No. 80 Refractory Cements and Plastics are now being exclusively distributed in the Chicago territory by Mayer & Oswald, Inc., 332 South LaSalle St., Chicago, Ill.

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Coatesville Boiler Works, Coatesville, Pa., recently acquired the Middletown, Pa., plant of the Standard Steel Car Company which comprises 40 acres of land and 500,000 sq ft. of floor area. This addition makes the capacity of the Coatesville Boiler Works the largest in America devoted exclusively to steel plate fabrication.

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General Refractories Company, Philadelphia, Pa., announces the opening of a new district office at Birmingham, Ala. This office is located in the Empire Building and is in charge of Walter S. Stapler.

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King Refractories Company, Buffalo, N. Y., has announced the appointment of Frederick H. Low as secretary and general manager. Mr. Low has been employed for the past several years in plant engineering work for the Ford Motor Company, at Detroit and is the son of Fred R. Low, editor emeritus of *Power.*

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The Fuller Lehigh Company, manufacturer of pulverized coal firing equipment and furnaces, has opened a new sales office in the Candler Building, Atlanta, Ga., with J. McC. Hill in charge.

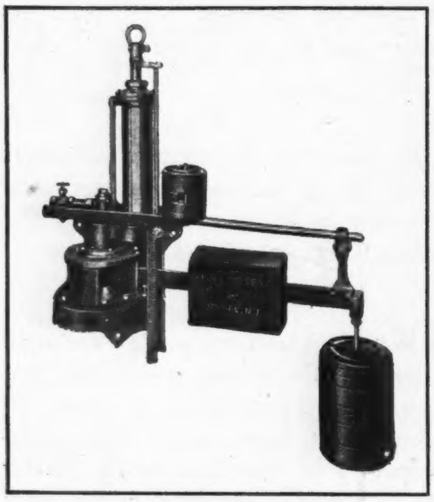
NEW EQUIPMENT

of interest to steam plant Engineers

Damper Regulator

An improvement in compensating type damper regulators has been announced by the Atlas Valve Company, 282 South St., Newark, N. J. This regulator is of novel design and was developed to provide sensitive and correct control of boiler dampers and to assure dependable operation.

The piston stem of the regulator, is connected with the boiler damper by a chain which runs over pulleys, so that vertical movements of the piston stem are translated into corresponding movements of the damper. The piston stem is operated in both directions by water pressure. The compensating feature comprises means for controlling the opening and closing of the valves which admit and discharge water from the two sides of the piston, and an auxiliary weight arm which is lengthened



and shortened automatically by the movement of the piston stem in a manner which prevents "hunting."

This new regulator will be sold under the trade name "No. 503 VICTOR Damper Regulator."

Automatic Boiler Feed

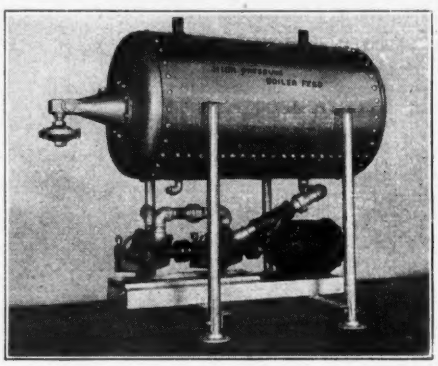
A simple device for feeding water to heating boilers has been developed by Mears-Kane-Ofeldt, 1903 East Hagert St., Philadelphia.

The M-K-O Automatic Boiler Feed is recommended for use where the operating steam pressure exceeds the city water pressure. It will feed pump water against boiler steam pressures up to 125 lb. so as to maintain a constant water level and will also return condensate to the boiler and supply make-up water as required.

No steam is used in operating the M-K-O Automatic Boiler Feed, so it makes no drain on the steam supply, and assures a constant water level without fluctuating-steam pressure.

These features coupled with complete automatic control relieve the boiler operator of feed water regulation by hand.

An electric switch mounted on a float chamber attached to the boiler makes contact as the float falls with the water level, starting the motor driven turbine pump. The pump feeds water until the rising float again throws the switch, breaking contact and stopping the motor. This method of control permits holding the water level limits within ½ inch.



The pump draws its water supply from a receiver tank vented to the atmosphere, into which all the condensation is returned by ordinary steam traps and to which make-up water is supplied, if required, by the float type feed water regulator built into one end.

The cycle of operation repeats as required without variation or attention, saving boiler fuel through saving condensation and eliminating feed water worries for the owner.

The M-K-O Automatic Boiler Feed can be used just as effectively for feeding and returning condensation to two or more boilers. When installed for two boilers, either boiler can be operated and fed independently of the other, or both at the same time.

Automatic Ratio Control

A ratio Control for the mixing of gases or liquids in any desired proportion has been announced by Bailey Meter Company, Cleveland, Ohio.

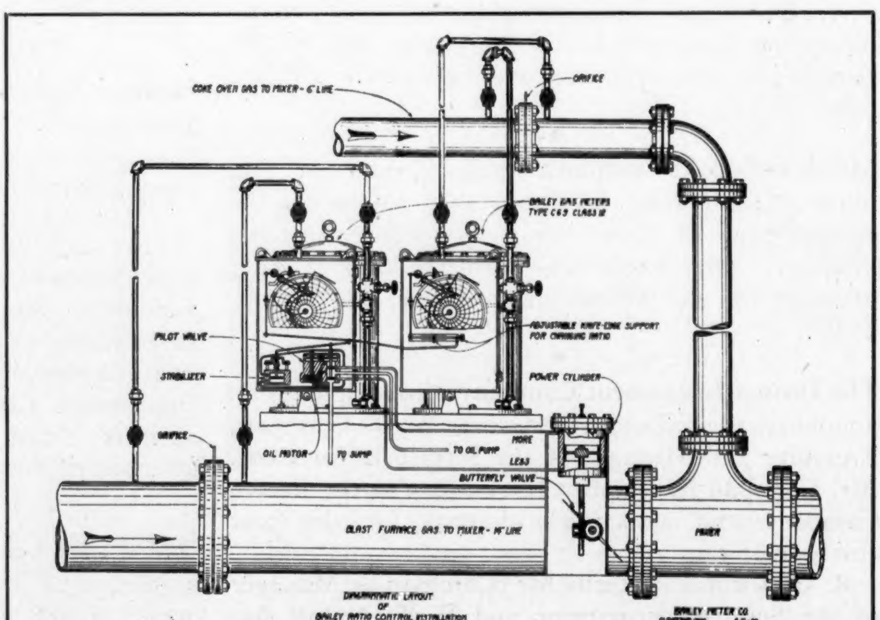
This control finds many applications in steel mills, oil refineries and process industries where mixtures of fluids must be definitely proportioned in a given ratio to secure the best results.

Air and fuel for combustion in industrial furnaces may be automatically proportioned by this control so as to secure maximum fuel economy.

An interesting example of the application of Bailey Ratio Control is in the automatic proportioning of blast furnace gas and coke oven gas. Blast furnace gas in itself is too lean and low in heating value to make a satisfactory fuel. Coke oven gas is too valuable to use by itself as a fuel but a mixture of two parts blast furnace gas to one part coke oven gas forms an ideal low cost fuel. This mixture is maintained by the Bailey Ratio Control.

As will be noted from the illustration at the bottom of this page, the control is operated by hydraulic pressure. Any change in the rate of flow in the coke oven gas line is immediately reflected by the Bailey Gas Meter connected to that line. The recording mechanism of this meter is connected through a system of linkage to a pilot valve, stabilizer and Bailey Gas Meter, which meter is in turn connected to the bast furnace gas line. If the flow of coke oven gas increases, the linkage moves so as to raise the pilot valve, allowing oil pressure to be applied to the "more" side of the power cylinder. At the same time, this movement opens the "less" side of the power cylinder to drain. The stabilizer tends to counteract the impulse given by the coke oven gas meter. This counteracting effect of the stabilizer is secured by connecting the "more" oil line to the upper chamber of the stabilizer and the "less" oil line to the lower chamber of the stabilizer. As pressure is applied to the "more" oil line and the "less" side is open to drain, the stabilizer piston moves downward and tends to return the pilot valve to a neutral position which in turn tends to equalize the pressures on either side of the stabilizer, causing it also to take a neutral position.

Any desired ratio between the two component parts of a mixture can be secured by selecting the proper ratios for the meter orifices. A further adjustment or change of ratio can be had by use of the ratio changing device which is simply a movable pivot supporting the linkage in the right hand meter casing.



REVIEW OF NEW TECHNICAL BOOKS

Any of the books reviewed on this page may be secured from In-Ce-Co Publishing Corporation, 200 Madison Avenue, New York

Experimental Mechanical Engineering

By Diederichs and Andrae

THE rapid expansion of the use of engineering instruments and the development of new methods of testing during the past twenty years have for some time called for a complete revision of the old text on these subjects. The present work is a completely rewritten text based upon the seventh edition of "Experimental Engineering" by R. C. Carpenter and H. Diederichs. The large amount of new material it has been found necessary to add has caused the work to be issued in two volumes. The present book, Volume 1, covers Engineering Instruments, their construction, use, and calibration. Volume II, to be issued in 1932, will cover the Testing of Power Plant Apparatus, which deals with the selection of engineering instruments and their applications to machines under test in order to attain properly the object of a given test. This second volume is based largely upon the Test Codes issued by the American Society of Mechanical Engineers.

The work is intended both for the student in engineering schools and for the practicing engineer. For the needs of the student the discussion on the theory involved in any given case is necessarily in many instances more extended than is needed for the engineer in practice. It is hoped, however, that the breadth of the field covered, the extended references to the sources of the information, and the degree to which practical constants are cited may make the work a satisfactory reference volume for the engineer in the field.

In order to describe adequately certain types of instruments, material furnished by manufacturers has been used. Referring to this in the foreword, the authors state that it was not possible to describe at length all the commercial forms that a given type of instrument may have taken, and the use of the product turned out by one manufacturer is not, therefore, to be construed as any special endorsement of the particular construction.

The work is divided into chapters, each of which is practically a treatise, complete in itself, of the topic or topics discussed. Following a discussion of engineering instruments, chapters are devoted to measurement of length and area; measurement of time and speed; measurement of pressure; measurement of temperature; measurement of work and power; the engine indicator; properties of gases and vapors; determination of moisture in steam; measurement of liquids, gases and vapors; fuel anaylsis and determination of heat value; exhaust and flue gas analysis; lubrication and the testing of lubricants.

The greatest expansion, in the way of added material over the previous edition, has taken place in the chapters on Speed Measurement, on the Engine Indicator, and on the Methods of Measuring Flow of Fluids. In the last field, particularly, a great deal of work has been done in the last ten or fifteen years, and a special effort has been made to bring this subject up-to-date.

The appendix includes vapor tables for steam, ammonia, sulphur dioxide and carbon dioxide; also vapor diagrams for the same gases and the A.S.M.E. test code for solid fuels.

This book is 6¼ by 9¼ overall and contains 1082 pages. Price $8.00.

Mechanical Engineers' Handbook

Lionel S. Marks, Editor-in-Chief, Third Edition

IT IS unnecessary to say a great deal about Marks' Mechanical Engineers' Handbook. The third edition of this great reference work has been thoroughly revised in al parts, bringing it up to date in both practice and theory.

Theoretical discussions have been strengthened to supply the mechanical engineer with the theory and data demanded by constantly increasing limits in sizes, pressures, temperatures, speeds and other conditioning factors. By calling in more contributors it has been possible to subdivide the sections more thoroughly and to give more specific treatment to various branches of mechanical engineering. Standards and practice have been brought up to 1930; physical data have been everywhere revised to incorporate the best current values.

The more important new sections include the following topics:—dimensional analysis—vibration problems—refractories—high-temperature carbonization of coal and gas making—low-temperature carbonization of coal—industrial combustion furnaces—electric industrial furnaces.

Many other sections of the book have been entirely rewritten and practically all sections have undergone extensive changes.

The new edition is bound in flexible covers, is 4½ by 7 overall and contains 2264 pages with over 1300 illustrations and diagrams. All sections are thumb indexed. Price $7.00.

NEW CATALOGS AND BULLETINS

Any of the following publications will be sent to you upon request. Address your request direct to the manufacturer and mention COMBUSTION Magazine

Boilers

Heine Longitudinal Drum Boilers are described in Catalog No. 52 recently reprinted. These boilers are of the box-header type and are adapted to either vertical or horizontal baffling. Twenty-eight illustrations are shown including line cuts of typical installation arrangements for various methods of firing. A double page spread is devoted to manufacturing facilities at the two Heine shops. 36 pages and cover, 8½ x 11—Combustion Engineering Corporation, 200 Madison Avenue, New York.

Cast Iron Storage Bunkers

Hahn Sectional Cast Iron Storage Bunkers are offered for storing ashes, coal, coke, sand, gravel and other materials of corrosive and abrasive nature. Designs with octagonal and oblong octagonal cross sections and having bottom discharge, are featured in a new illustrated folder. 6 pages, 8½ x 11—Treadwell Engineering Co., Hahn Division, Easton, Pa.

Combustion Control

L. & N. Metered Combustion Control applied to boilers in a plant equipped with two-speed inlet vane or dampered fans is described in Bulletin No. L-661. A typical plant consisting of six boilers operating at 650 lb. pressure and having a maximum steam output of 500,000 lb. per hour each, is assumed. The principles of operation are defined and the functioning of each element in the control system is described in detail. 8 pages, 8 x 10½—Leeds & Northrup Company, Philadelphia, Pa.

Conveyors

Palmer-Bee Conveyors are illustrated and described in a new two-color catalog No. 52. While overhead conveyors for general industrial use are featured, a complete line of coal and ash conveyors and skip hoists is available. Brief references is made to Palmer-Bee Herringbone Speed Reducers and to the facilities of this company for fabricating structural steel for buildings and supporting structures. More than one hundred illustrations show various types of conveyors and their applications. 80 pages and cover, 8½ x 11—Palmer Bee Company, Detroit, Mich.

Flow Meters

Instruction Book No. 214 covers the installation and care of Brown Electric Flow Meters. The principles of operation are described and detail instructions are given for the location, installation, adjustment and care of meters. Fifty-three illustrations are presented covering details of design, application arrangements, wiring diagrams and drilling dimensions. Numerous tables of correction factors are also included. 40 pages and cover, 8 x 10½—The Brown Instrument Company, Wayne and Roberts Aves., Philadelphia, Pa.

Gas Burners

A new Atmospheric Gas Burner for boilers is presented in Bulletin No. 306 recently issued. The burner nozzle is of refractory tile shaped to form a Venturi tube so the gas jet produces an injector effect and the operation is not entirely dependent upon boiler draft. This feature eliminates puffing and vibrating. Louver type air shutters are adapted to either hand operation or automatic control. The burners will operate satisfactorily at any gas pressure from 2 ounces to 40 lb. per sq. in. A table of sizes and capacities is included. 4 pages, 8½ x 11—The Denver Fire Clay Co., Denver, Colo.

Heat Resisting Paints

CE-CO Heat Resisting Paints and Asphaltum Base Paints are described in Bulletin No. 10. A wide range of paints is offered covering standard colors in addition to black and white and adapted to use at high temperatures—up to 1000 deg. fahr. or even higher under certain conditions. These paints are recommended for stacks, breechings, furnaces and cupolas and are both temperature and corrosion resisting. CE-CO Plastic Stack Lining for stacks, flues and cinder catchers is also described. 20 pages, 8 x 10½—Cheesman-Elliot Co., 639 Kent Ave., Brooklyn, N.Y.

Industrial Furnaces

A new bulletin describes the range of work of the recently formed Swindell-Dressler Corporation which represents 80 years' experience in the design and building of industrial furnaces. The scope of work of the new company includes, industrial furnaces, electric furnaces, tunnel kilns, gas burners, and plant supervision, reports and engineering service. 24 pages and cover. 6 x 9—Swindell-Dressler Corporation, P.O. Box 1753, Pittsburgh, Pa.

Plastic Refractory

"Cutting Furnace Costs" is the title of a new booklet describing the advantages and application of Plibrico, a plastic, putty-like refractory. The booklet is printed in two colors throughout and includes numerous illustrations of installations and recommended applications. Nine pages are devoted to setting drawings which show furnace arrangements for various types of boilers and methods of firing. 40 pages and cover, 8½ x 11—Plibrico Jointless Firebrick Company, 1800 Kingsbury Street, Chicago.

Recording Pyrometer

An interesting description of the New Uehling Self-Contact Potentiometer Pyrometer appears in Bulletin No. 301. This bulletin includes a discussion of pyrometry in general and discloses the many features of the Self-Contact Potentiometer Pyrometer the novelty of which lies primarily in the fact that no depressor bars, cam mechanisms, or continuously operating motors are utilized. The galvanometer which actuates the pen mechanism does not form part of the recording gage. The instrument makes a continuous line record over any desired range. The temperature is also automatically indicated over a two-foot illuminated scale. 12 pages, 8½ x 11—Uehling Instrument Co., 4 Vesper St., Paterson, N. J.

Small Stoker

"Profits from Stoker Firing," is the title of a new bulletin describing the Modern Coal Burner, an underfeed stoker for burning bituminous coal under small boilers. Coal is fed from the stoker hopper to the retort by means of a feed-screw which is driven by an electric motor. The drive mechanism provides three operating speeds and is enclosed in a sealed housing so that the gears run in oil. A separate motor-driven fan unit supplies forced draft. Numerous illustrations show details of construction and advantages of design. 18 pages, 8½ x 11—The Modern Coal Burner Co., 3733 Lincoln Ave., Chicago, Ill.

Smoke Indicator

The Eclipse Smoke and Combustion Indicator is a periscopic device which projects onto a ground glass, within view of the fireman, a beam of light which has passed across the stream of gas in the boiler flue. Correct combustion is usually indicated by a thin grey haze which is readily recognized on the indicator. Dense smoke obstructs the passage of the light and the glass is dark, while the entire absence of smoke and the resulting bright reflection, shows large excess air. A description of the indicator and instructions for installation are available. 6 pages, 8½ x 11—Boiler Room Improvement Company, 4057 Van Buren Street, Chicago, Ill.

Underfeed Stokers

Detroit Single Retort Stokers are described in Bulletin No. 201, just published. This bulletin illustrates the double control of the feed of coal and its distribution in the furnace, methods of distributing air to fuel bed, automatic regulation with both steam engine and variable speed motor drive, and other features. Charts and records of fuel savings and other operating economies are shown, together with line prints illustrating the application of these stokers to many different types of boilers. 32 pages and cover, 8½ -x11—Detroit Stoker Company, 1227 General Motors Bldg., Detroit, Mich.

Water Softener

The Graver Zeolite Water Softener is presented in Bulletin No. 509. The Zeolite method of water softening is also known as the exchange method because when water containing hardening salts is passed through a bed of zeolite the lime and magnesia are absorbed from the water and replaced by the soda which the zeolite contains. No sludge or precipitates are formed in the process. When the exchange of sodium for calcium and magnesium has reached its limit, the process is reversed and a strong salt solution is passed through the zeolite to restore its original properties. 16 pages, 8½ x 11—Graver Tank & Manufacturing Company, East Chicago, Indiana.

NOTICE

Manufacturers are requested to send copies of their new catalogs and bulletins for review on this page. Address copies of your new literature to

COMBUSTION
200 Madison Ave., New York

WALSH & WEIDNER

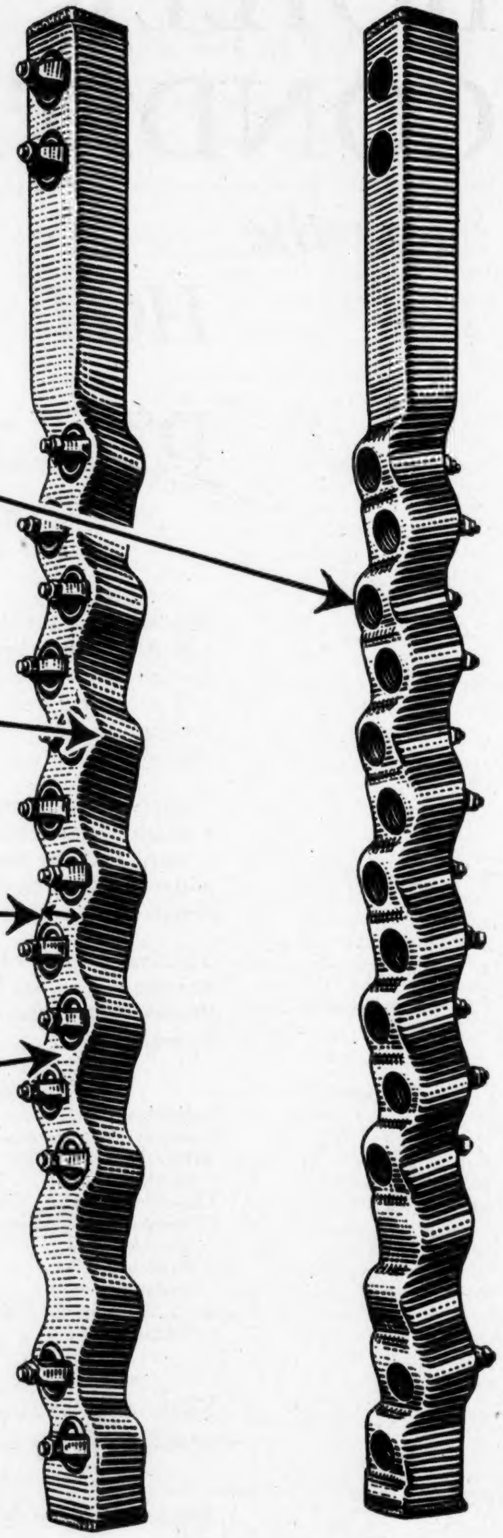
Seamless Forged Steel Headers

NO LENGTHWISE WELDS

LARGE FILLETS

DIMENSIONS DO NOT VARY AND HEADERS ARE INTERCHANGEABLE

FORGED SOLID FROM BEST GRADE WROUGHT STEEL SEAMLESS TUBING



Walsh & Weidner Sectional Header Boilers are made in sizes up to 40,000 square feet of heating surface.

COMBUSTION ENGINEERING CORPORATION

200 Madison Avenue **New York, N. Y.**

BOILERS - AIR PREHEATERS - STOKERS - PULVERIZED FUEL EQUIPMENT - WATER-COOLED FURNACES

National Power Show
New York, December 1 to 6

UNDER the leadership of I. E. Moultrop, Chairman, plans for the Ninth National Exposition of Power and Mechanical Engineering to be held December 1 to 6, in Grand Central Palace, New York City, are well under way.

The exhibits this year will be of particular interest as rapid developments in the power field have resulted in the design of much new equipment and in improvements in almost all lines of power apparatus.

When the first New York Power Show opened its doors nine years ago, pulverized coal for power use was in its infancy and opinion was divided as to its possibilities. Today, in the public utility field alone over 7,000,000 tons of coal is burned in pulverized form each year. When the first Exhibit was held, steam pressures of 300 to 400 lb. were spoken of as high pressures. This year, pressures of 1400 to 1800 lb. and higher are being discussed. Boiler designs for these pressures and for the high capacity, high rating service typical of modern steam plant operation will be represented at the 1930 Power Show.

Mechanical stokers of various types will be shown as well as water-cooled furnaces, air preheaters, economizers, superheaters and all manner of power plant auxiliary equipment.

The Advisory Committee includes the following: Homer Adams, Past President American Society Heating and Ventilating Engineers; N. A. Carle, Manager Pacific Electric Mfg. Co.; H. D. Edwards, President American Society Refrigerating Engineers; Fred Felderman, Past National President National Association Power Engineers; Vincent M. Frost, Chairman Power Division American Society Mechanical Engineers; L. A. Harding, President American Society Heating and Ventilating Engineers; C. F. Hirshfield, Chief Research Department, Detroit Edison Company; O. P. Hood, Chief Mechanical Engineer, U. S. Bureau of Mines; John H. Lawrence, President Thomas E. Murray Co.; Fred R. Low, Past President American Society Mechanical Engineers; David Moffett Myers, Consulting Engineer; Fred W. Payne, Co-Manager Exposition; Charles Piez, President American Society Mechanical Engineers; Joseph W. Roe, Chairman Professional Division American Society Mechanical Engineers; Charles F. Roth, Co-Manager Exposition, and W. A. Jones, President National Electric Light Association.

The growth in interest in this field is indicated by the fact that there were only 105 exhibits at the first show while at this time with the 1930 show three months away more than 400 exhibitors have already taken space and are actively planning to make this the best year that the National Power Show has had.

Since the National Power Show will not be held in 1931, this forthcoming Exposition will draw better exhibits and a record attendance of visitors.

Fifth Midwest Power Engineering Conference

At Chicago—February 10 to 13, 1931

Twenty representatives of the foremost engineering societies in the Power field met recently in Chicago to outline plans and arrangements for the FIFTH MIDWEST POWER ENGINEERING CONFERENCE. Past meetings of the Conference have been held in Chicago since 1925 and the next meeting promises to be of greater importance than any preceeding it.

The work of carrying on the four day meeting scheduled for February 10 to 13, 1931, inclusive, at Chicago will be in the hands of the following officers recently elected: President, H. W. Fuller of Byllesby Engineering & Management Corp.; Vice-Presidents, C. C. Whittier, Chicago, W. M. White, Milwaukee, Wis., Charles S. Gladden, Detroit, Mich., Paul Doty, St. Paul, Minn., John Hunter, St. Louis, Mo., Prof. A. C. Willard, Urbana, Ill.; Secretary, G. E. Pfisterer; Treasurer, K. A. Auty. Headquarters have been established at 308 West Washington Street, Chicago, Illinois.

The Conference will be sponsored by the local sections and regional divisions of the following:

American Institute of Electrical Engineers
American Society of Civil Engineers
American Society of Mechanical Engineers
American Institute of Mining Engineers
American Society of Refrigerating Engineers
National Safety Council
National Electric Light Association
Western Society of Engineers

Several committees have already started work for the four day meeting and a canvass is being made to secure the latest ideas and most modern applications of Power and its allied subjects. The rapid progress in this field, and in the field of Physics so closely allied, will allow a presentation of subjects of national interest. Leading authorities in the heating, refrigerating, mining, electrical and mechanical fields will be asked for expressions of the progress of Power application in these fields, thereby giving a comprehensive survey of generation distribution and utilization in a large way.

The real purpose of the POWER CONFERENCE at Chicago is announced as an effort to coordinate the various problems in the Power field so as to give a broader understanding of applications in many different fields of use. The part of Power in our daily life and in special work will find expression in the diversified papers contemplated. Original papers are being arranged with ample opportunity for discussion by leading authorities. It is expected that this Conference will represent one of the most outstanding of its kind ever held in this country.

Boiler, Stoker and Pulverized Fuel Equipment Sales

Total figures to July 1, as reported to the Department of Commerce by the leading manufacturers in each industry

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Boiler Sales

	Total 6 mo. 1930		Total 6 mo. 1929		June, 1929		June, 1930	
	No.	Sq. ft.	No.	Sq. ft.	No.	Sq. ft.	No.	Sq. ft.
Water tube	601	3,163,274	851	4,960,457	179	934,042	130	568,940
Horizontal return tubular	479	660,093	673	903,222	138	188,372	104	137,747

Mechanical Stoker Sales

Year and Month	Total		Type of Boiler			
			Fire-tube		Water-tube	
	No.	HP.	No.	HP.	No.	HP.
1929						
January	97	43,392	36	5,835	61	36,557
February	80	31,554	26	3,933	54	27,621
March	117	42,432	42	6,369	75	36,063
April	141	48,749	41	6,508	100	42,241
May	174	60,772	64	9,951	110	50,821
June	203	67,322	82	13,323	121	53,999
Total (6 mo.)	812	293,221	291	45,919	521	247,302
Total (year)	1,716	599,585	706	102,515	1,010	497,070
1930						
January	53	13,198	24	2,872	29	10,326
February	73	22,648	26	3,732	47	18,916
March	89	32,403	45	6,128	44	26,275
April	108	35,903	46	6,984	62	28,919
May	96	31,956	41	5,703	55	26,253
June	145	48,467	70	13,080	75	35,387
Total (6 mo.)	564	184,575	252	38,499	312	145,976

Pulverized Fuel Equipment Sales

Year and Month	Central System			Unit System		
	No. of Pulverizers	Total Rated capacity in tons of coal per hour	Total Rated hp. of boilers equipped	No. of Pulverizers	Total Rated capacity in tons of coal per hour	Total Rated hp. of boilers equipped
	For Installation Under Water-Tube Boilers					
1930						
January	1	6	1,600	52	565	59,742
February	2	20	3,000	29	175	23,305
March	2	50	6,414	16	33	9,995
April	....	....	....	31	139	37,993
May	3	80	11,360	30	196	22,625
June	1	6	802	15	28	7,146
Total (6 mo.)	9	162	23,176	173	1,136	160,806
	For Installation Under Fire-Tube Boilers					
January	....	....	....	6	35	965
February	....	....	....	2	13	305
March	....	....	....	3	3	450
April	....	....	....	3	3	780
May	....	....	....	....	....	....
June	....	....	....	....	....	....
Total (6 mo.)	....	....	....	14	54	2,500